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MANUFACTURING COST STUDY OF SELECTED GAS TURBINE AUTOMOBILE ENGINE CONCEPTS

E. S. Wright, L. E. Greenwald, W. R. Davison

United Aircraft Research Laboratories
East Hartford, Connecticut 06108

AUGUST 1971

FINAL REPORT

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OFFICE OF AIR PROGRAMS

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United Aircraft Research Laboratories

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UNITED AIRCRAFT CORPORATION
EAST HARTFORD, CONNECTICUT

Report K-971017-4

Manufacturing Cost Study of Selected Gas Turbine Automobile Engine Concepts

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SUMMARY

A feasibility study of several selected gas turbine automobile engine concepts has been conducted to estimate the probable manufacturing cost of the engines in quantities of 100,000 and 1,000,000 units annually; to perform system analyses of the candidate engines by comparing probable vehicle performance, fuel economy, manufacturing costs, and emissions; and to recommend future research and development goals.

Four candidate gas turbine engines were designed in sufficient detail to permit the estimation of engine performance and probable manufacturing costs, namely a conventional low-pressure-ratio (4:1), regenerated, free-turbine engine representative of past and current automotive designs, and three engines based on the use of advanced centrifugal compressors and radial turbines extrapolated from demonstrations by United Aircraft of Canada Limited. One of the advanced engines utilizes a regenerator, but the other two are high-pressure-ratio (10 to 12:1), simple-cycle engines, one of them a single-shaft and the other a free-turbine version.

Engine performance was estimated on the basis of applicable component test data, using computer programs; vehicle performance and emissions data were estimated with the use of a computer simulation program developed especially for this purpose. Manufacturing costs were estimated by appropriate combinations of detailed production methods analysis and estimates from potential vendors. Particular attention was devoted to those parts whose probable costs represent a significant portion of the projected manufacturing costs of the total engine. A critical-materials analysis was conducted to evaluate the possibility of exceeding the available supply at production volumes of 10 million units per year. A research and development program designed to implement the most attractive gas turbine concept is outlined.

CONCLUSIONS

1. A simple-cycle gas turbine engine design has been found feasible for automobiles. This engine is based on United Aircraft Corporation power plant component technology, the most significant factor being the ability to obtain high compression ratios in a small, single-stage compressor at high efficiencies. The resulting engine is the most attractive of the gas turbine engines studied because of its predicted low manufacturing cost and low NO_x emissions level coupled with adequate fuel economy.

2. An automobile powered by this simple-cycle gas turbine has excellent potential towards meeting US Government emission standards for 1976. Predictions based on past simple-cycle combustor technology for aircraft engines indicate that NO_x emissions reductions of up to 40% may be necessary to comply with 1976 NO_x emissions standards; however this reduction, as well as the required reductions in UHC and CO, is believed achievable and is an objective of combustor development programs now under way with support from the Office of Air Programs.

3. The manufacturing costs predicted for the single-shaft version of this simple-cycle gas turbine are in the range associated with future low-emission Otto-cycle automobile engines, provided that manufacturing volume is comparable and that the appropriate manufacturing processes are implemented. However, transmissions which would permit satisfactory operation with a single-shaft gas turbine require further performance and cost definition. The manufacturing costs predicted for regenerated gas turbines (similar to earlier automotive gas turbine designs) are substantially higher than those for the simple-cycle single-shaft engine.

4. The predicted costs of acquiring, owning, and operating an automobile powered by this simple-cycle single-shaft gas turbine engine are competitive with present experience (as defined by US Department of Transportation reports). A regenerated single-shaft engine based on the same level of aerodynamic technology is also competitive, since its somewhat higher acquisition costs are balanced by its better fuel economy. The low-pressure-ratio regenerated free-turbine engine studied exceeds present costs by at least 10%, but is competitive with the future 1976 Otto cycle-engined automobile assumed for this report.

5. Based on computer simulations, the performance of standard-sized six-passenger automobiles powered by the 150-hp gas turbine engines considered in this study should exceed the startup, acceleration, and grade velocity specifications of the Office of Air Programs (OAP) , provided the automobiles are equipped with suitable transmissions. (Further optimization studies are under way, with OAP support, to define engine size and type more closely.)

6. It is not likely that the widespread substitution of these simple-cycle gas turbines for Otto-cycle engines will result in critical material demands affecting the availability of the constituent metals. With particular reference to nickel, the maximum projected consumption at production levels of 10 million engines annually would total only 15% of present US consumption, and less than 0.2% annually of estimated world reserves. A more critical supply problem exists with regard to cobalt (33% of present US consumption), but the requirement for cobalt is less than 10% of the requirement for nickel, and the general interchangeability of cobalt- and nickel-based refractory alloys offer many tradeoffs should either metal be in critical supply.

7. The aerodynamic and thermodynamic technology required for a suitable simple-cycle gas turbine engine has been sufficiently demonstrated to permit a high degree of optimism in the attainment of the engine performance predictions; however, turbine and compressor demonstrations at design sizes are necessary to confirm scaling assumptions, clearance effects, low-speed operation characteristics, and to provide data for optimization investigations. Development programs are also required for low-emission combustors (currently in progress), bearings, controls, reduction gearing, and transmissions.

8. The overall feasibility of the simple-cycle single-shaft gas turbine automobile engine studied in this report appears sufficiently attractive to justify full-scale demonstration in automobiles. Development of the engine, suitable transmissions, and pertinent accessories to the extent necessary for such a demonstration is desirable, since if this demonstration program is successful, the widespread adoption of these engines will be a practical means of drastically reducing vehicular-caused air pollution.

RECOMMENDATIONS

It is recommended that a two-phase development program be conducted to further develop the gas turbine technology and manufacturing processes necessary to implement the advanced-technology, simple-cycle engine concept, the first phase of which would include demonstration of operating engines in automobiles. This work can be initiated concurrently with optimization studies which examine all potential gas turbine engine configurations and designs for the automobile application. The demonstration program would consist of a logical sequence of design and development tasks for components, systems, and manufacturing processes. The initial research and development tasks relative to engine components would include development of combustors (already under way) and of compressors and turbines. Concurrent work should be performed on gearbox and bearing components, as well as on transmission systems, and on manufacturing techniques which include high-ductility forging processes. The first-phase program would culminate in the construction and demonstration of several engines - initially on test stands and ultimately in automobiles.

The second phase of the program would commence following a successful first-phase demonstration. It would involve the far more extensive effort required to design, develop, and test engines intended for actual large-scale production.

INTRODUCTION

The favorable emissions characteristics of gas turbines, when contrasted with those associated with present-day spark-ignition automotive engines, have prompted renewed interest in the study of this type of power plant for automobile propulsion. Advancing gas turbine technology, as provided by continuing development in the aircraft engine field, offers a means of achieving high thermodynamic efficiency in combination with low emissions, and recent advances in manufacturing techniques, in conjunction with a high-volume market, offer promise of production costs which are consistent with automotive production cost requirements.

Accordingly, a manufacturing cost study of automobile gas turbines (No. EHS 70-115) was sponsored by the National Air Pollution Control Administration (NAPCA) of the Department of Health, Education, and Welfare* at United Aircraft Corporation. The major objectives of the study were to evaluate the relative feasibility of several types of automotive gas turbine engines, including a conventional regenerated engine, an advanced regenerated engine, and free-turbine and single-shaft versions of a simple-cycle gas turbine. In the establishment of feasibility, emphasis was placed on the documentation of manufacturing costs for the selected engines. Additional tasks were to estimate the emissions and performance, including fuel economy, of automobiles powered with the selected gas turbines, and to compare them with conventional automobiles powered by spark-ignition engines. A final task was to develop recommendations concerning the direction of OAP-sponsored research and development in automotive gas turbines, including the identification of those areas in gas turbine technology and manufacturing which require additional research.

The United Aircraft Research Laboratories (UARL) provided program management and performed the systems analysis tasks relating to the evaluation of vehicle performance and operation, the cost analysis, and the identification of recommendations for future gas turbine development. United Aircraft of Canada Limited (UACL) performed cycle analyses and prepared component designs, engine arrangements, and design details for the simple-cycle engines selected for study. Regenerated engine designs were synthesized by Homer J. Wood and Associates** as a consultant to UARL.

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* Since the award of this contract, the NAPCA has been superseded by the Office of Air Programs (OAP) of the Environmental Protection Agency (EPA). Consequently, reference to the sponsoring agency in this report will be by the designation OAP.

** A mechanical engineering consultant firm specializing in small turbomachinery, located in Sherman Oaks, California, also known as PowerDynetics Incorporated.

In the performance of the manufacturing cost analysis, UARL was assisted by the production and industrial engineering staffs at UACL and by several cooperating potential vendors, including Eaton Corporation, Misco Division of Howmet Industries, the Borg-Warner Research Laboratories, Holley Carburetor Division of Colt Industries, the Hamilton Standard and Pratt & Whitney Divisions of United Aircraft, Corning Glass, TRW, and SKF Industries.

DESIGNS

General

Approach

Gas turbines have often been considered for use in automobiles for a number of reasons. However, a major impediment to their adoption in this field has been the expected high manufacturing cost of typical existing designs (see, for example, Ref. 1)*. This apparent obstacle appears to be due to two characteristics typical of these designs, namely (1) their complexity (caused by cycle and control sophistication generally believed to be necessary to achieve competitive fuel consumption and performance), and (2) their relatively high content of high-cost refractory-alloy materials.

A previous industrial engine study conducted by United Aircraft Corporation has indicated that a design approach based on technology developments which have been demonstrated by United Aircraft, coupled with mass production techniques, could lead to impressive reductions in expected manufacturing costs. This study led to the speculation that, in the case of the automobile application (with its relatively modest efficiency requirements compared with industrial gas turbine and diesel engines), this approach might represent a design breakthrough which would result in the desired low manufacturing costs. Specifically, it appeared that a simple-cycle gas turbine engine might prove practical.

The technology base for this breakthrough is the result of advanced aerospace components developed and demonstrated at United Aircraft of Canada Limited (UACL). UACL has carried out a series of research programs aimed at providing advanced components for small gas turbines for aircraft since 1961. Much of the research effort has been concerned with advanced centrifugal compressors and radial turbines and with sophisticated analytical stressing and mechanical design innovations required to make them feasible. Some of the results of these programs have been published in Refs. 2, 3, 4, 5, 6, 7, 8, and 9. The development work on these components points the way toward achieving high compression ratios in a single-stage compressor at high efficiencies; this capability offers the promise of eliminating the customary regenerator while maintaining acceptable fuel economy, thus leading to a simple-cycle automobile gas turbine characterized by both advanced performance and low manufacturing cost potential.

---

* References are located on page 165.

Selection and Identification of Engine Designs

The study was not intended to serve as a comprehensive review of all possible gas turbine engine cycles and configurations for the automobile application*. Therefore only a few of the most interesting concepts were selected for consideration in order to evaluate the simple-cycle approach. One such engine was included by OAP in their original study request, namely, a 4:1-pressure ratio, regenerated engine of free-turbine configuration (RFT-4) similar to designs announced and/or tested for automotive applications by General Motors, Ford, Chrysler, Williams Research, etc. This engine type might be considered as the "conventional" automotive gas turbine, since most gas turbine research, development, and production activity within the automotive industry has been devoted to engines of this type. The configurations based on the use of the high-pressure-ratio components mentioned above were selected in order to provide comparisons in performance and manufacturing cost for alternative advanced-technology designs. These designs were for (1) a single-shaft version of the simple-cycle engine (SSS-12), (2) a free-turbine version of the simple-cycle engine (SFT-12), and (3) a regenerated single-shaft version which differed from the conventional gas turbine primarily by virtue of operation at higher pressure ratios (RSS-7).

A form of identification which preserves the essential differences (cycle, output, and pressure ratio) among the gas turbine engine types is used throughout this report. This identification of the engines listed in the above paragraph is derived as follows:

SSS-12: Simple-Cycle Single-Shaft, 12:1 pressure ratio

SFT-12: Simple-Cycle Free-Turbine, 12:1 pressure ratio

RSS-7: Regenerated-Cycle Single-Shaft, 7:1 pressure ratio

RFT-4: Regenerated-Cycle Free-Turbine, 4:1 pressure ratio

In addition to the gas turbine engines studied, comparisons are made in the report with a conventional 1970 automobile spark-ignition Otto-cycle V-8 engine of approximately 300 in.^{3} displacement, and its 1976 counterpart, which presumably has been modified to meet 1976 Federal emissions standards. These engines are referred to as OC-70 (Otto-cycle 1970) and OC-76 (Otto-cycle 1976), respectively.

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* Studies to define the optimum configuration for a low-emission automobile gas turbine engine are currently under way with OAP sponsorship.

It should be noted that the greatest study effort has been concentrated on the SSS-12 engine, since the preliminary calculations made prior to the instigation of this study indicated that its manufacturing cost would be the lowest of all of the candidate gas turbine engines.

Study Assumptions

The study was based on the following major assumptions (or ground rules):

1. All designs considered were specified to meet 1976-78 dates for start of production. The 1976 date selection was subsequently justified by the passage of clean air legislation which requires automobile manufacturers to meet stringent Federal emissions standards by that year; however, it might not be possible to produce the engines studied at a rate large enough to significantly affect air pollution by that date. In the case of all gas turbine engines considered, meeting the 1976 date for start of production will be a difficult (but not impossible) task which will require some overlap in development and tooling programs. Engine performance was estimated on the premise that the predicted performance need not necessarily be met the first year of production, but only within one or two years following, thus providing the best estimate of the potential usefulness of these engines. In other words, it was assumed that the engine could be put in production by 1976, but that the life, reliability, and efficiency goals postulated might not be obtained immediately.

2. The production rates considered for the manufacturing cost analysis were specified as 100,000 and 1,000,000 units annually by the terms of the study contract. The 100,000-unit level is not representative of current automobile engine production rates in the United States, but might represent a production run of one type of engine on an assembly line with capacity for producing much larger annual volumes. The 1,000,000-unit figure is believed to be more representative of automobile engine production rates. Within this study it was decided, nevertheless, that the 100,000-unit annual production level would be costed as the total production level of a given production line. This rule leads to somewhat higher costs when compared with automobile experience, but it nevertheless yields results of interest, since innovations in the automobile industry traditionally take several years to reach high-volume production.

3. In order to insure the maximum credibility for the study, it was decided that only gas turbine technology which had actually been rig-tested and demonstrated would be considered, with modest extrapolations

of demonstrated performance made, in many cases, for the best estimate of 1976-78 performance. Therefore, approaches which might have great promise for a low-cost gas turbine engine but which, in the contractor's opinion, were not sufficiently demonstrated (such as air bearings or extensive use of ceramic parts other than as a regenerator heat exchanger matrix) were not considered in the evaluations; consequently, the potential of second-generation engines was not evaluated, although they can be expected to have both improved efficiency and lower costs than those considered here. This exclusion was a direct result of the requirement that the engines be capable of limited production in 1976. The major exception which was taken to this approach was for the design of the regenerator for the RSS-7 engine. In this case, a proprietary design was used which has not yet been rig-tested. It was thought useful to include this type in order to obtain a cost comparison between the ceramic-matrix type of rotary regenerator (RFT-4) and a design based on more conventional metal technology (RSS-7), but it must be emphasized that both of the regenerated engines (RSS-7 and RFT-4) could have been designed with either type of regenerator.

4. As in the case of gas turbine technology, it was required that all manufacturing processes specified be demonstrated or in use today, but not necessarily at the production volumes required. Automobile production ground rules were applied to these processes, so that highly automated production lines with multiple-head machines and automatic transfer lines were considered. The least degree of compliance to the demonstrated-production-method rule is the consideration of the use of high-ductility forging for the forming of rotors on the advanced-technology engines. This technique has been used to produce rotor disks for JT12 aircraft engines, which have been tested successfully, but has yet to be demonstrated for the production of completely bladed rotors. The use of this process is discussed more fully on page 117.

5. The direct manufacturing costs, as defined herein, are considered to exclude the costs for tooling, factory space, etc., which are obviously important parts of the overall costs of producing any given engine. These costs depend on a number of extraneous factors, and both the actual costs and the manner in which they are allocated (prorated) vary widely among different manufacturers and situations. On the basis of rather cursory considerations by UACL and UARL, it appears that the costs of capital equipment required to manufacture the simple-cycle single-shaft gas turbine engine should not exceed 10% of the engine manufacturing cost on a fully amortized basis at the 1,000,000-unit/year rate. The corresponding expenses incurred by vendors are, however, included in the costs of vendor-furnished parts (as would normally be the case for purchased parts).

6. In all cases, the designs have been executed only to the degree necessary to permit realistic costing, but not to the degree necessary to manufacture an operating engine. This procedure is justified by the assumption that the details of the exact shapes of the various parts, which may be crucial in determining performance, are generally not critical in estimating costs during the feasibility analysis stage, provided materials and tolerances are specified.

Engine Types

SSS-12

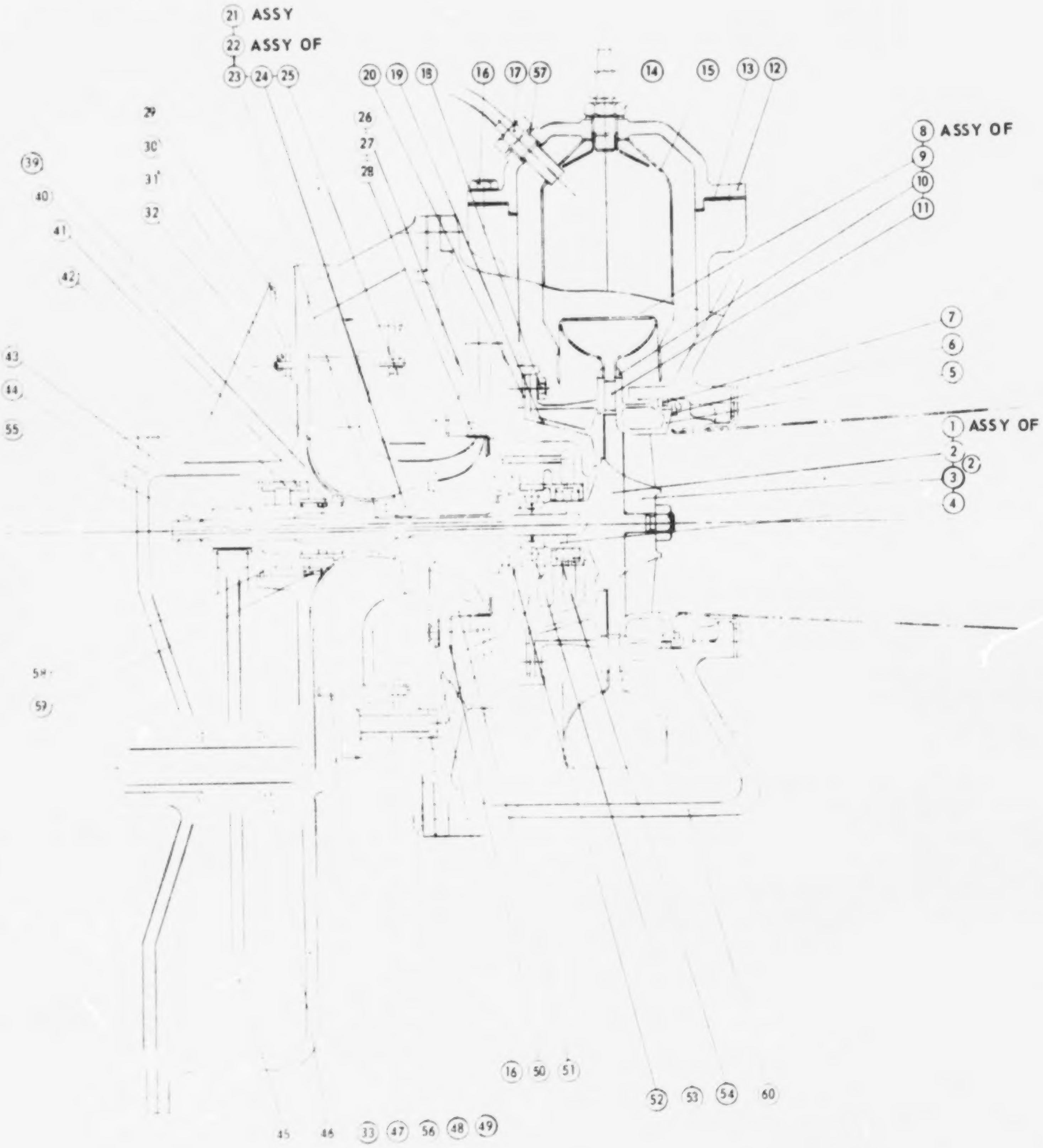
The layout of the simple-cycle single-shaft engine (SSS-12) is shown in Fig. 1. This engine is based completely on advanced-technology radial components and is very simple in concept. An end view of the design and a full-size sketch of the gas path are shown in Figs. 2 and 3, respectively. As can be seen, compression is provided by a single-stage centrifugal impeller pipe-diffuser combination of advanced concept. Heat is added by a conventional single-can combustor. Expansion is through an advanced-concept single-stage radial turbine providing both compressor drive and output power. A picture of a full-size mockup of this engine is shown in Fig. 4. The mechanical design of the two simple-cycle engines (SSS-12 and SFT-12) is intended to complement the advanced aerodynamic components. This has been done by maintaining the maximum simplicity compatible with very low mechanical and leakage losses and the necessary quality of the critical aerodynamic components. Although this design philosophy has frequently resulted in materials or components which may be considered "aircraft quality", particularly in the rotors and bearings, this choice was not made from a lack of appreciation of the requirements for low-cost production. Rather, it was made with an understanding of the minimum acceptable component quality which permits the high-pressure-ratio, high-efficiency simple-cycle gas turbine to compete with traditional automotive power units. It will become apparent later that the resulting engines have attractive manufacturing costs.

The standard-day design point conditions for the SSS-12 engine are shown in Table I, (together with similar information for all of the gas turbine engines studied). The design point turbine inlet temperature was derived from an analysis of operating cycles for the engine in 3,000 hours of typical automobile use and can be summarized as equivalent to 100 hours at maximum turbine blade metal temperature and 10,000 start-run-stop cycles. These life values are dependent on the type of transmission used (infinitely variable or stepped ratio) and the engine running line chosen, as this affects the variation of turbine temperature with power and the duration of transient temperatures. A preliminary evaluation of the turbine rotor was conducted and satisfactory life and adequate burst speed margin can be expected for a rotor forged from superalloy turbine materials for this cycle.

FIG. 1

LAYOUT, SIMPLE-CYCLE SINGLE-SHAFT 12:1 ENGINE (SSS-12)

NOTE: PART NUMBERS ARE KEYED TO TABLE X


21 ASSY
22 ASSY OF
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16
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57
14
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29
30
31
32
26
27
28
39
40
41
42
8 ASSY OF
9
10
11
7
6
5
43
44
55
1 ASSY OF
2
2
3
4
58
59
16
50
51
52
53
54
60
45
46
33
47
56
48
49

FIG. 2

END VIEW, EXHAUST END, SSS-12

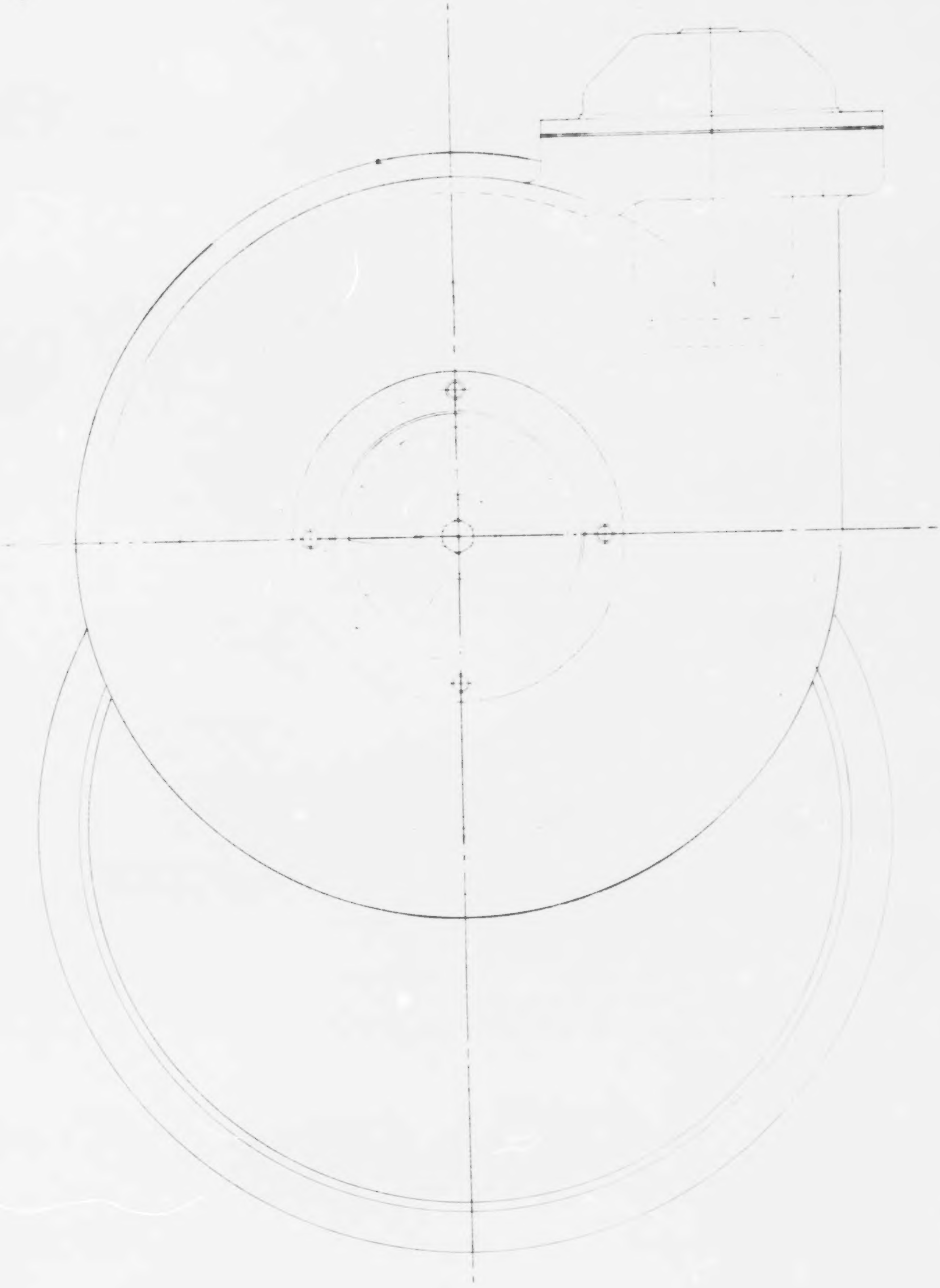
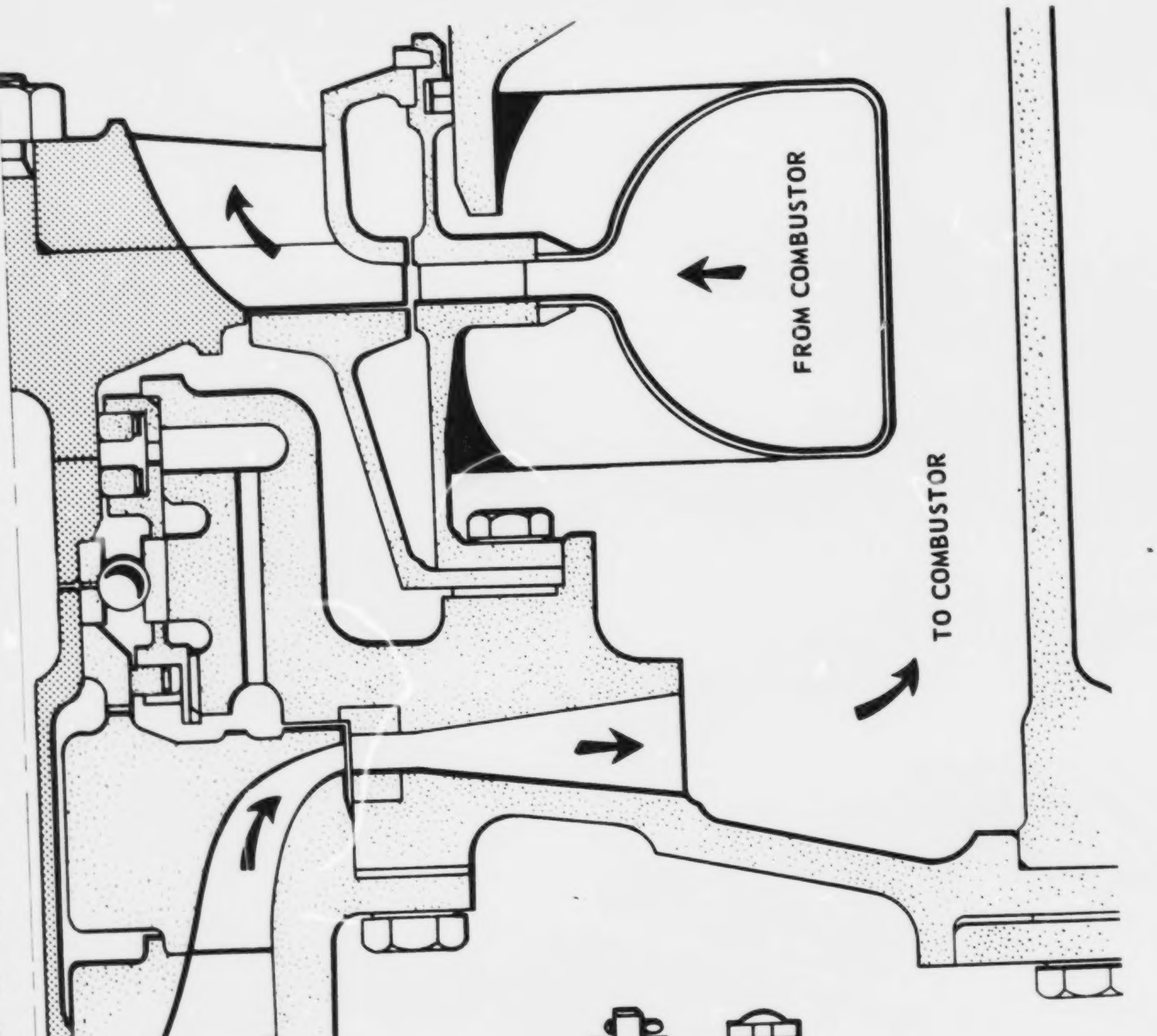


FIG 3


FROM COMBUSTOR
TO COMBUSTOR

SKETCH OF GAS PATH, FULL SIZE, SSS-12

K-971017-4

(FULL SIZE)

FIG. 4

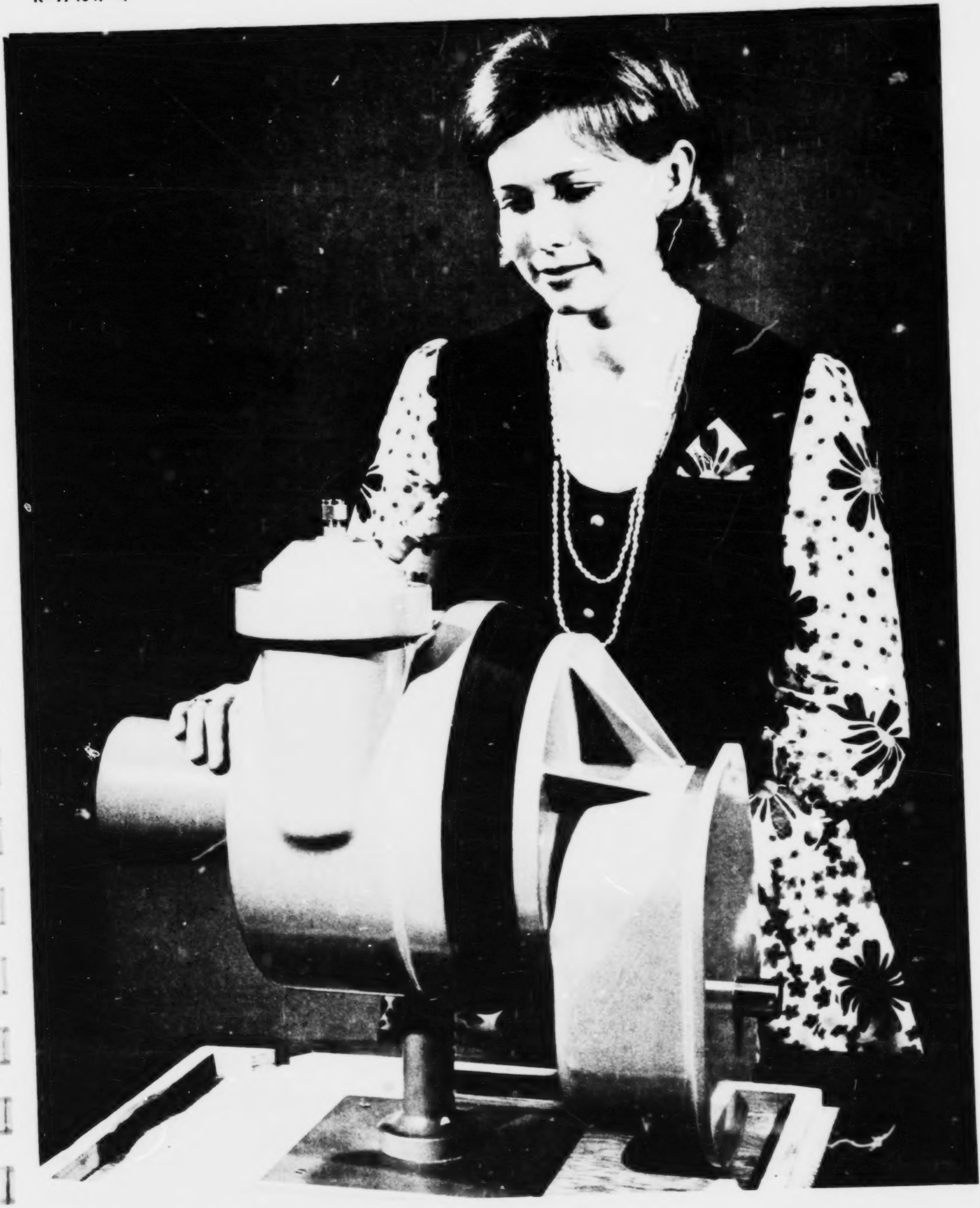


TABLE I

ENGINE DESIGN POINTS - STANDARD DAY

	SSS-12	SFT-12	RSS-7	RFT-4
Compressor Shaft Speed - rpm	130 K*	130 K*	112 K	66.5 K
Power Turbine Shaft Speed - rpm	N/A	90 K	N/A	71.3 K
Output Drive Speed - rpm	13 K	13 K	12 K	6 K
Airflow - lb/sec	1.06	1.06	1.25	1.67
Inlet Losses - %	0.9	0.9	1	1
Compressor Pressure Ratio - To Unity	11.1*	11.1*	7.2	4.14
Compressor Efficiency (Total to Static) - %	78.5	78.5	80.0	78.0
Combustor $\Delta P/P$ - %	2	2	3	5
Combustor Efficiency - %	98	98	99	99
Turbine Inlet Temperature - F	1950*	1950*	2000	1850
Turbine Pressure Ratio - To Unity	10.1	10.1	6.5	3.75
Turbine Efficiency (Total to Total) - %	88.3	88.3	88	85
Exhaust Losses - %	6.3	6.3	1	1
Parasitic Losses - hp	3.8	3.8	2	2
Engine Gearbox Loss - hp	4.2	4.2	6	6
Net Output - hp	150	150	163**	166**
Fuel Flow - lb/hr	76.2***	76.2***	73.4	88
BSFC - lb/hp-hr	0.51	0.51	0.45	0.53
Regenerator Effectiveness - %	N/A	N/A	49	62
Carryover and Seal Losses - %	N/A	N/A	4	4
Regenerator Power Loss - % of Heat Transferred	N/A	N/A	4	4

---

* Maximum (hot day) values are 135 Krpm, 12:1 pressure ratio, and 2000 deg.

** Assumed to be derated for 150-hp operation.

*** Calculated in October 1970 corresponding to the above conditions. The fuel flow estimates used in this report (Figs. 9 and 12) were derived from calculations made in May-June 1971, as discussed on p. 22.

Component Descriptions

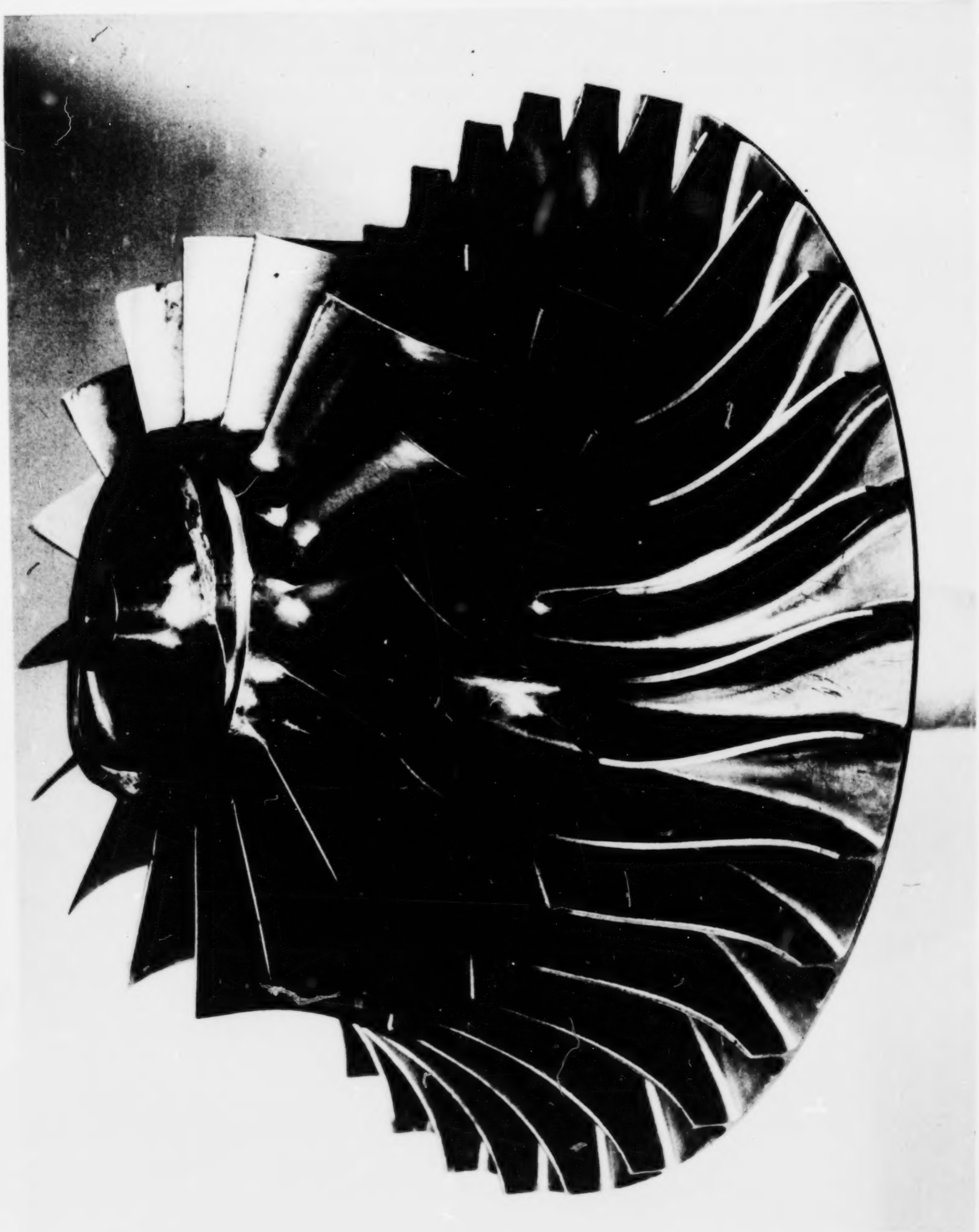
The single most advanced component in this engine, and the key to the whole concept, is the high-pressure-ratio single-stage compressor. The various types of compressors considered for the higher-pressure-ratio engines during this study are the result of an advanced-concept centrifugal-compressor research program jointly funded by the Canadian Defense Research Board and UACL, early results from which have been fairly widely reported, e.g., Ref. 6. A picture of a typical high-pressure-ratio centrifugal impeller and of a typical pipe diffuser, both of which were developed under this program, are shown in Figs. 5 and 6, respectively.

Tests for the 12:1 compressor were originally accomplished with hardware designed for an airflow of about 15 lb/sec, but with a throttled intake, so that the Reynolds number is equivalent to a compressor flow of 2.1 lb/sec at atmospheric inlet pressure, as compared with the design value of 1.06 lb/sec for the SSS-12. Recent additional high-pressure-ratio tests were conducted on smaller compressors designed for airflows of 2 to 2.5 lb/sec, but tested at flows of 1.2 to 1.4 lb/sec and maximum speeds of 85,000 to 94,000 rpm.

All compressor maps used to determine performance for the advanced-technology engines in this study were derived from these test results, with appropriate corrections for Reynolds number and hardware scaling and, in some cases, assuming modest expected efficiency improvements as a result of planned development programs. The compressor map originally used for the SSS-12 engine design was prepared in August 1970. It predicted the rig-demonstrated performance expected to be achieved by the end of 1975. This map was scaled from the larger-flow compressor test results and is shown in Fig. 7. At pressure ratios of 4, 7, and 11, corresponding to peak thermodynamic powers (standard day) of 40, 70, and 170 shp, the maximum compressor efficiency predicted for 1975 was 72.2, 73.0, and 79.3% total-to-static, respectively.

By November 1970, Reynolds number-corrected values of 75.0, 77.0 and 72.2, respectively, had been obtained on the large rig. By March 1971, corresponding efficiencies of 73.8, 75.7, and 73.8% had been obtained on the smaller compressors running with very nearly the correct-size hardware and, due to intake throttling, at Reynolds numbers lower than the expected engine values. Therefore the 1975 predicted efficiency values for part-load operation have already been exceeded in tests, a factor which is significant since the great majority of automobile engine fuel consumption occurs under these conditions.

FIG. 5



TYPICAL UACL ADVANCED CENTRIFUGAL IMPELLER

FIG. 6

TYPICAL UACL PIPE DIFFUSER

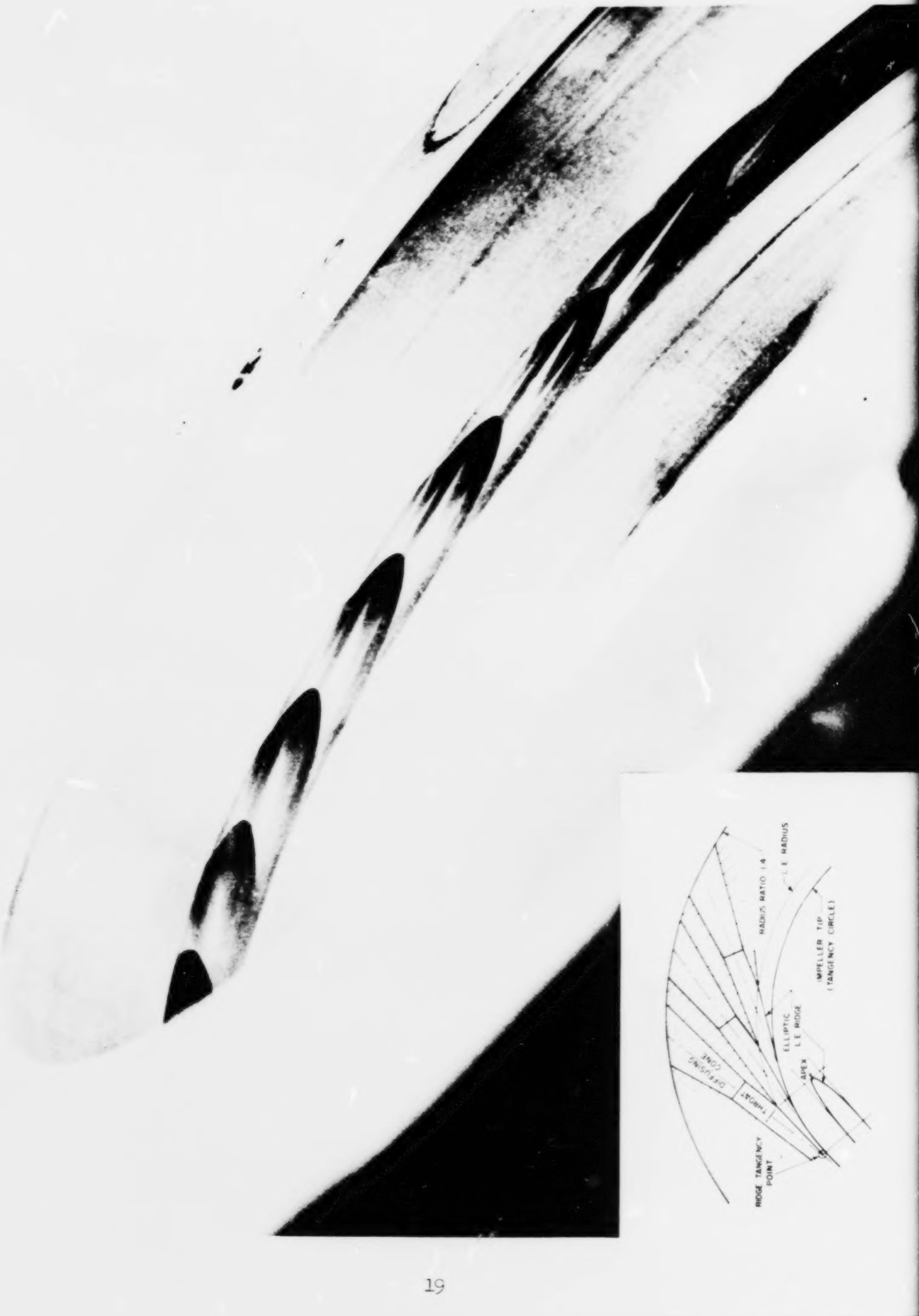
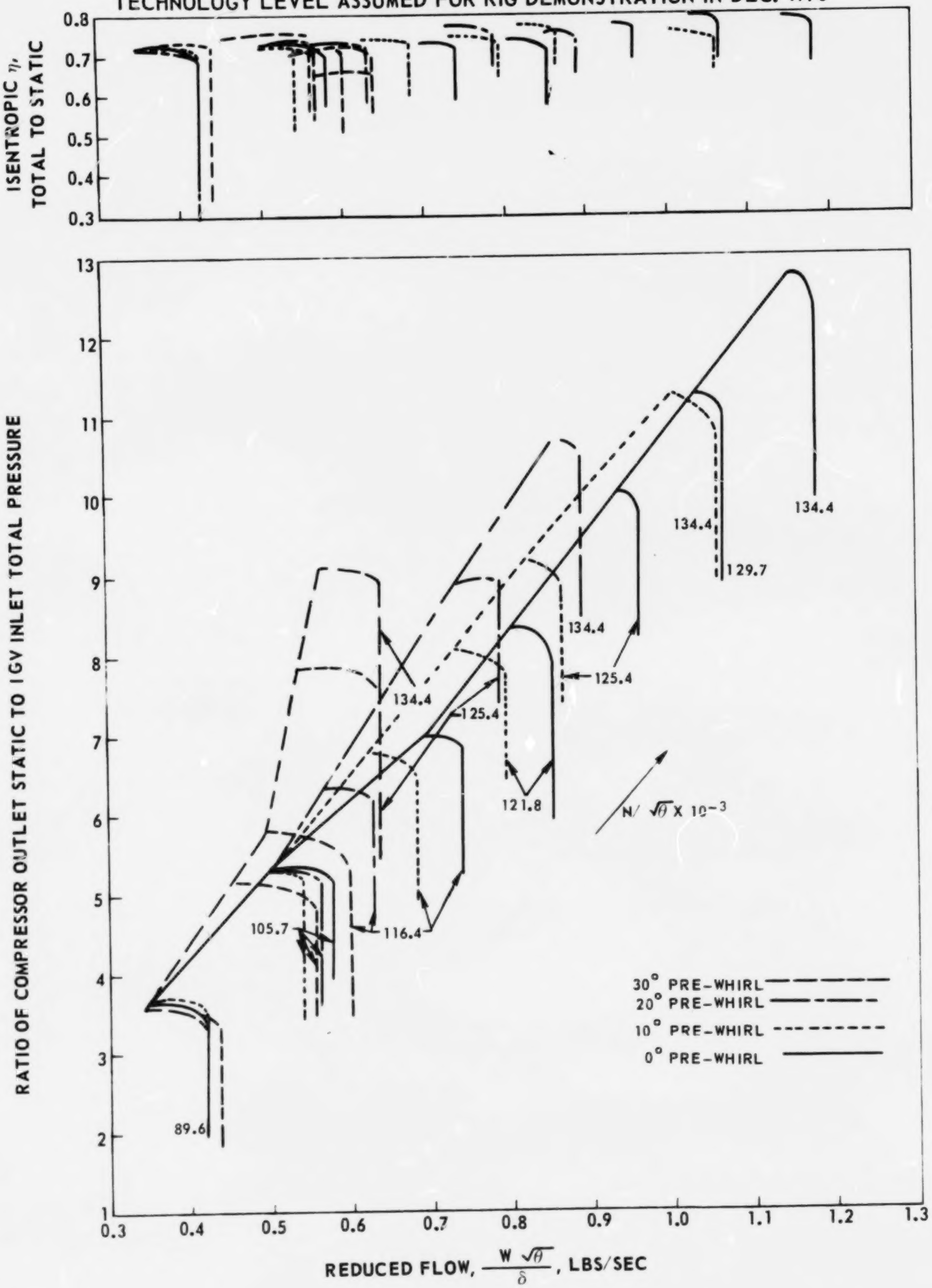

RADIUS RATIO 1.4
L.E. RADIUS
IMPELLER TIP (TANGENCY CIRCLE)
ELLIPTIC L.E. RIDGE
APEX
DIFFUSING CONE
THROAT
RIDGE TANGENCY POINT

FIG. 7

COMPRESSOR PERFORMANCE ESTIMATE, SSS-12, SFT-12

TECHNOLOGY LEVEL ASSUMED FOR RIG DEMONSTRATION IN DEC. 1975



Variable inlet guide vanes are shown in Fig. 1, but they can possibly be eliminated on the final production engine design as a result of compressor improvements discussed above. They do not add significantly to the overall cost of the engine while providing a powerful means of optimizing engine operation without the cost, performance penalties, and mechanical unreliability of variable-geometry hot parts, such as power turbine nozzles.

The combustor is a single-can design selected for inherent reliability and low potential exhaust emissions. The combustion efficiency was assumed to be 98%, which is a conventional value. Actual combustion efficiency is expected to be 100%, but the small loss indicated is a conservative allowance for radiant heat losses up to the turbine exit. The combustor static-to-total pressure drop of 2%* of combustor inlet pressure is approximately twice the value of current production PT6 engines, and hence is also considered a conservative value.

The radial turbine is an especially challenging component for the single-shaft version of the nonregenerative gas turbine. The design efficiency for this component was estimated as 88.5% total-to-total and 80% total-to-static. These figures are derived from UACL test data showing total-to-total efficiencies of 90% achieved at 7:1 pressure ratio under throttled conditions with a test Reynolds number 1.86 times that for the study engine. Analysis conducted to date indicates that this efficiency level will hold up to pressure ratios of 10 to 12:1 but further testing is needed to verify these predictions.

The exhaust is discharged to the rear through a high-efficiency diffuser. Intake and exhaust pressure losses ($\Delta P/P$) have been assumed to be 0.9% and 6.3%, respectively; this assumption implies a careful installation. As a measure of performance sensitivity to these factors, it has been determined that increasing the intake loss to 1.4% and exhaust loss to 7.3% would reduce net horsepower from 150 to 147 and increase typical duty-cycle fuel consumption by 0.6%.

The power output is to the front (cold end) of the engine and the engine maximum mechanical speed is 135,000 rpm. The reduction gear which has been drawn and costed for the single-shaft engine consists of a single-stage spur of 10:1 reduction. This selection was based on preliminary information regarding transmission input requirements which are now known to be premature, and it is more likely that a two-stage spur gear set would be needed to provide output speeds of from 3600 to 10,000 rpm, depending on the specific transmission selected. (The cost penalty for this variation would be minimal.) The pitch line velocity for the first stage pinion would be unchanged at 30,000 feet per minute (fpm) in either case. This high value is within modern practice since UACL has considerable experience with gears running at 24,000 fpm on current production engines and is running compressor test rigs at 40,000 fpm.

---

* 3% total-to-total

The accessories (not shown on this layout) would be located on or near the output casing, and would be belt or gear-driven in accordance with present automobile practice.

Both plain bearings and anti-friction bearings were considered for this engine. Production turbochargers are available today with low-cost plain bearings running in excess of 100,000 rpm, but calculations concerning conventional plain bearings indicated excessive friction power losses. Although these losses can probably be minimized by more careful attention to design, there is inconclusive evidence to suggest that the required low cost could still be retained. Greater confidence (based on reasonable extrapolation of production engine and test rig experience) can be placed on power loss and life calculations for anti-friction bearings and these were therefore specified. Although it remains to be proven that the resultant bearings can be produced at the required low cost, the preliminary estimates received from a potential bearing vendor are encouraging (see Table 10 on page 120.)

Running clearances on these small turbomachinery components are obviously of great importance. The short stiff structures locating the rotor and shrouds on this engine and the good location provided by the centrally mounted thrust bearing permit close control of these clearances. Analysis of the part tolerances shows that axial and radial clearances between shrouds and rotors for both compressor and turbine will be maintained within the range .007 to .013, for the parts as specified and costed. The effects on engine performance of these clearances is adequately accounted for in the engine performance estimates.

The engine is 17 inches long, 19 inches high, and 14 inches wide, and weighs approximately 120 lb without accessories (160 lb with). The use of a two-stage reduction gearbox would reduce these dimensions further.

Optimization and Revised Fuel Flow Predictions

The design effort on simple-cycle engines for automobile applications is an extremely active program at UACL, and review, optimization, and revision of all engine performance estimates are continually under way in order to reflect both the latest aerodynamic test data and the latest estimates on installation and mechanical losses. The original SSS-12 design was "frozen" in October 1970, based on design studies conducted at UACL during the previous three months, and the costing and the original vehicle performance estimates were prepared at UARL based on this design. An opportunity was provided to redo the original vehicle performance estimates during June 1971 (the results of which are described elsewhere in this report), and for these purposes the latest fuel flow estimates derived from UACL optimization studies conducted during May and June 1971 (but not as a part of this study) were used. In general, the optimization work has proceeded in the direction of matching the compressor to moderately lower pressure ratios to increase its efficiency at part power, where almost all the fuel is consumed. This procedure gives lower efficiency at the higher pressure ratios, thus reducing engine power modestly or requiring a small increase in engine size, and the actual technique used is to increase the diffuser throat area. For example, results from the small test compressor modified in this regard have shown efficiencies of 77.8, 78.3, and 73.0% at pressure ratios of 4, 7, and 9.5, respectively. The compressor map, Fig. 8, which was used to predict the fuel consumption for the simple-cycle engines shows some of the effects of matching to moderately lower pressure ratio. Like Fig. 7, it also represents predicted test performance for the end of 1975.

The fuel flow currently predicted for engines of the SSS-12 type is shown in Fig. 9. A simple IGV schedule of 30° at low speed is included. Although these estimates reflect an engine optimized for better overall fuel consumption, the actual fuel flow predicted is higher than those for the original calculations. The reason for this is that higher installation losses were included, and an arbitrary 2% overall increase in fuel flow was added as a contingency allowance to account for unforeseen development or production problems. This contingency allowance is equivalent to a reduction in compressor efficiency of 2-1/2%, or in turbine efficiency of 1-1/2%.

The revised fuel flow predictions for the SSS-12 type engine, shown in Fig. 9, assume that the engine develops 150 hp at full power. The full-power standard-day design point conditions for the revised engine are listed below where they differ from the original design point conditions listed in Table I.

Shaft Speed - 124.2 K rpm
Airflow - 1.16 lb/sec
Inlet Loss - 1.0%
Compressor PR - 10.33
Compressor η - 75.0
T.I.T. - 2000 F

Turbine PR - 9.0
Exhaust Loss - 9.6%
Engine Gearbox Loss - 5.0 hp
Net Output - 151.8 hp
Fuel Flow - 86.3 lb/hr
Bsfc - 0.57 lb/hphr

It will be noted that the combination of lower pressure ratio, lower compressor efficiency, greater exhaust losses, and 2% contingency allowance increase full power sfc by 11-1/2% over that shown in Table I for the original SSS-12 design.

Although the design airflow of the above engine has increased by almost 10% (1.06 lb/sec to 1.16 lb/sec), the corresponding changes in the engine manufacturing cost are expected to be less than 4%, and size changes of this magnitude are within the uncertainty estimates for the manufacturing cost predictions reported. It should be further noted that different models of the same physically sized engine would probably be matched to high pressure ratios for high performance or heavy-duty use and to lower pressure ratios for better low-power fuel consumption at the expense of peak power. If this procedure increased the total sales volume for the engine, unit manufacturing costs for each engine would probably decrease due to higher total engine production volume.

In view of the above, it is felt valid to use the original SSS-12 design for the manufacturing cost estimates, while using the revised SSS-12 engine fuel flow estimates for the automobile performance results. This procedure provides the best assessment of overall feasibility possible at this time, although all results reported herein are subject to further changes as optimization studies proceed.

FIG. 8

K-971017-4

TYPICAL HIGH-PRESSURE-RATIO COMPRESSOR PERFORMANCE PREDICTIONS

NOMINAL EXIT MACH NO. = 0.12
COMBINED 30°, 0° INLET SWIRL MAP
(30° SWIRL – 100% SPEED INTERPOLATED)

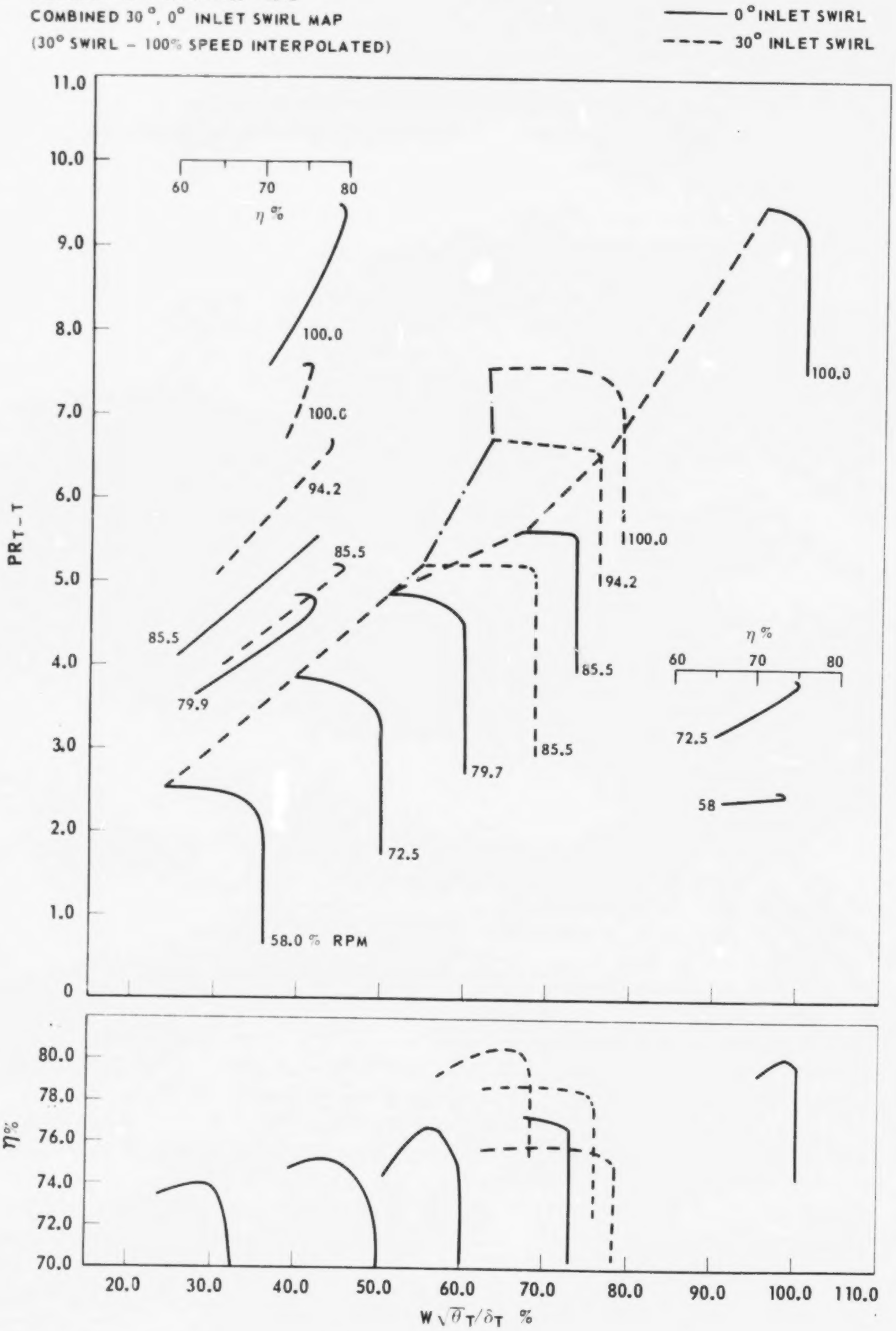
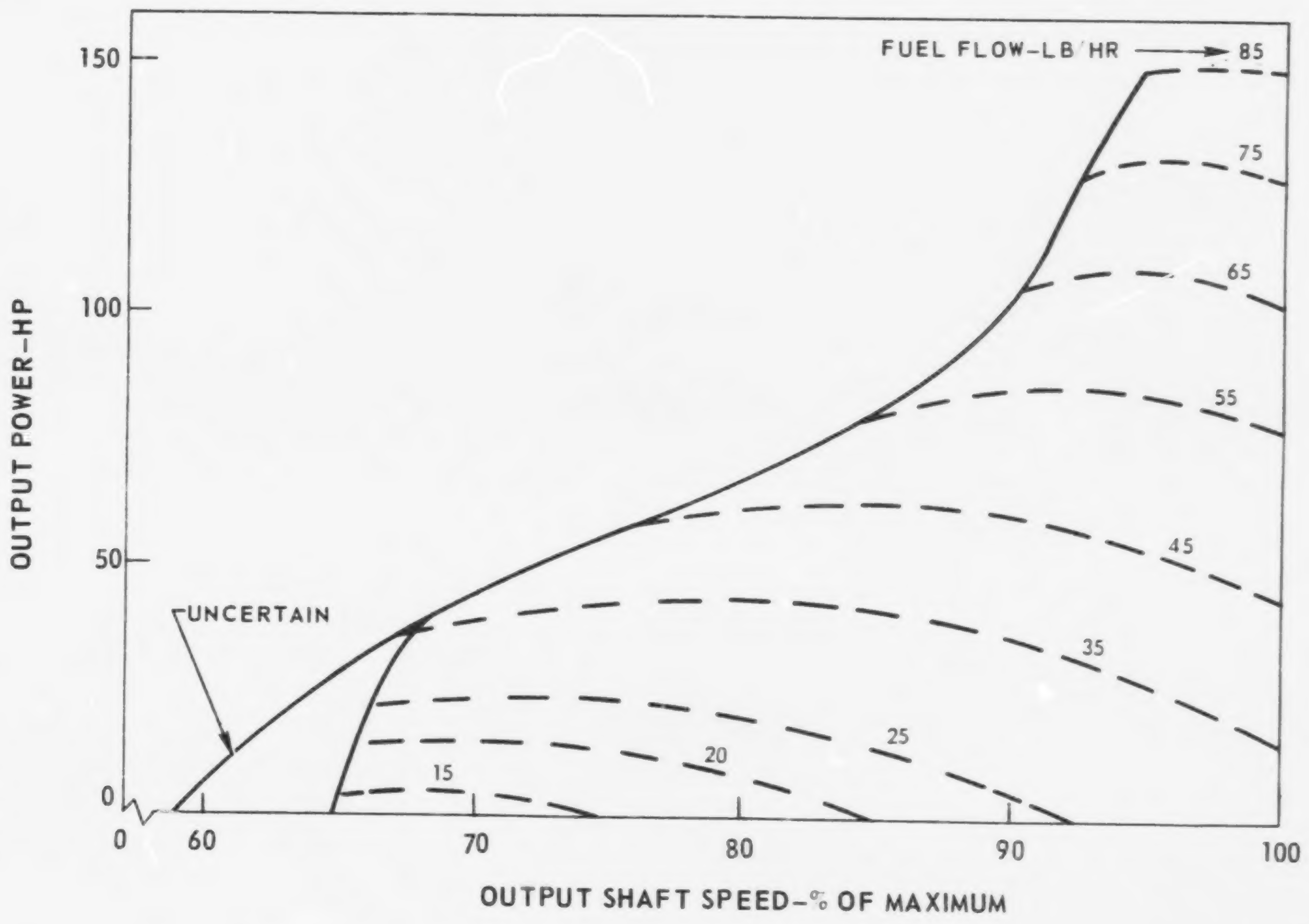


FIG. 9

ENGINE PERFORMANCE ESTIMATES, SSS-12

- COMPOSITE OF OPTIMUM INLET GUIDE VANE SETTINGS
- 5 % INSTALLATION LOSSES INCLUDED
- STANDARD DAY



SFT-12

The layout of the simple-cycle free-turbine engine designed as one alternative to the single-shaft engine previously described is shown in Fig. 10. A full-size sketch of the gas path is shown in Fig. 11. As can be seen, the addition of an extra turbine stage complicates the design considerably by adding another turbine wheel, a set of turbine nozzles, a bearing shaft, and an additional power output train. The primary output is to the rear (hot end) of the engine, with an additional output at the front of the engine for accessory drives. The compressor design is identical to that of the single-shaft engine described above, as is the combustor and the variable inlet guide vanes (which probably cannot be dispensed with in this engine). The radial turbine, which drives the compressor only, is somewhat smaller, although otherwise similar in design. The flow from this turbine is fed to a set of fixed axial turbine vanes and a single-stage axial power turbine which comprise the final expansion stage.

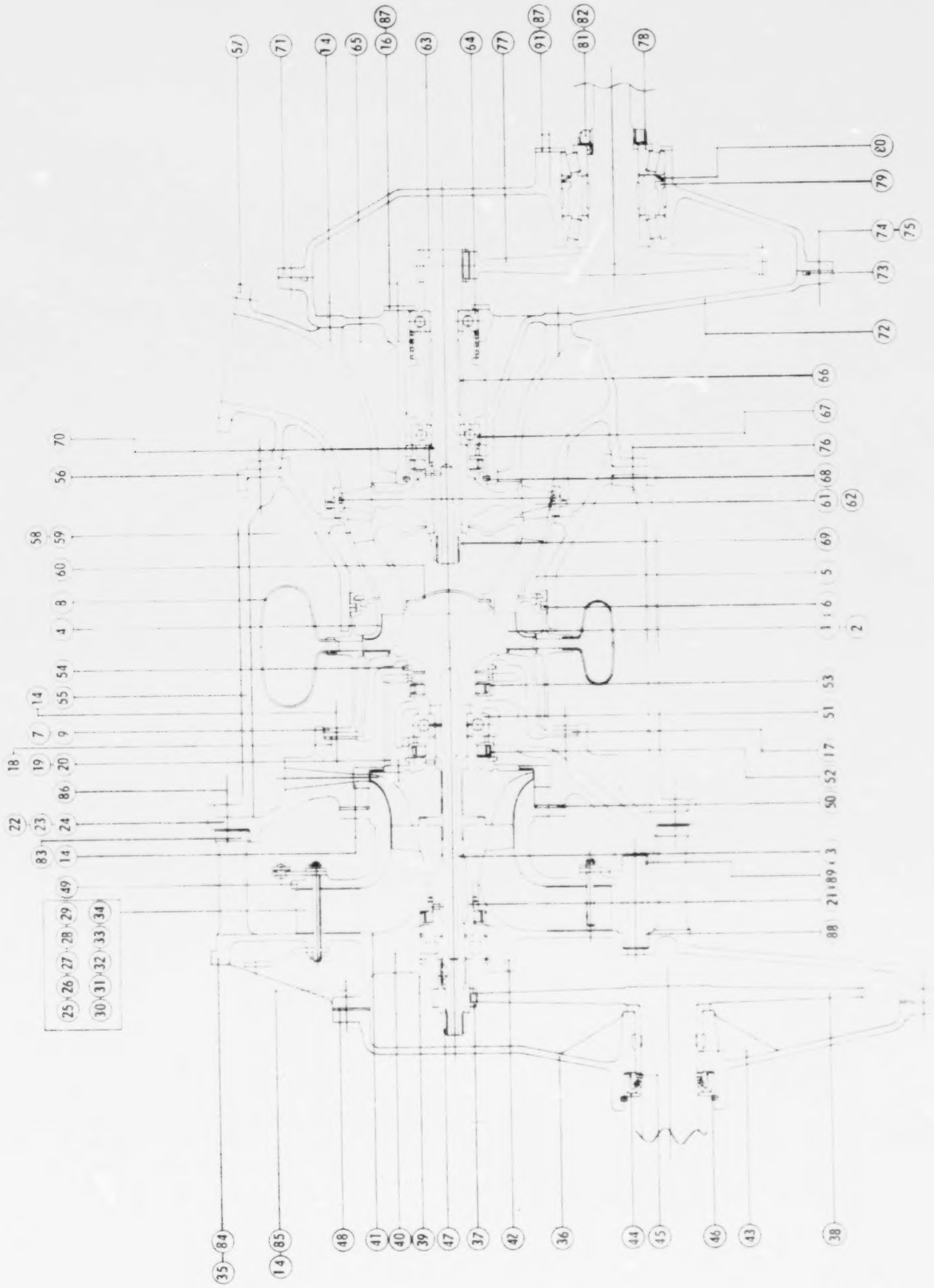
Rotational speeds for the engine are 135,000 rpm for the gas generator shaft and 90,000 rpm for the power turbine. As in the previous design, rolling contact bearings are shown, together with carbon seals, for minimum technical risk and low power losses. Exhaust is collected at the rear of the engine and delivered to a high-efficiency diffuser.

The design point conditions of this engine (Table I) are assumed to be identical with those of the SSS-12 wherever applicable. In an optimized design, it is probable that the smaller temperature drop across the free-turbine engine's radial turbine would result in higher metal temperatures in this component,requiring a reduction of 50° to 100° (F) in the turbine inlet temperature. However, the effect on efficiency of this drop would be compensated for by the expected slightly higher efficiency of the two-stage turbine configuration.

The axial power turbine on the high-pressure-ratio small engine presents a very difficult design problem. An extensive optimization among disc stress levels, cycle efficiency, and costs would be necessary for a complete operational design. Time and budget limitations did not allow this; consequently the design produced for this study must be considered representative for costing purposes only. The power turbine as drawn, with a bored disc, is fatigue-limited at the bore if made from one piece of forged Udimet 700. Blade stresses are satisfactory as the temperature is too low for creep to be a problem. The possibility of casting an integral wheel has been investigated. A solid, unbored disc is required and the fatigue life of the disc would then depend on the hub material properties that could be obtained from a casting development program. Although the wheel was designed to be shrouded, manufacturing cost can be lowered by casting it without the shroud at some expense in performance.

LAYOUT, SIMPLE-CYCLE FREE-TURBINE 12:1 ENGINE (SFT-12)

NOTE: PART NUMBERS ARE KEYED TO TABLE XI


57
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91-87
81-82
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74-75
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68
61
62
69
5
1-6
2
53
51
17
52
50
3
21-89
88
70
56
58
59
60
8
4
54
14
55
7
9
18
19
20
86
22
23
24
83
14
49
25-26-27-28-29
30-31-32-33-34
35-84
14-85
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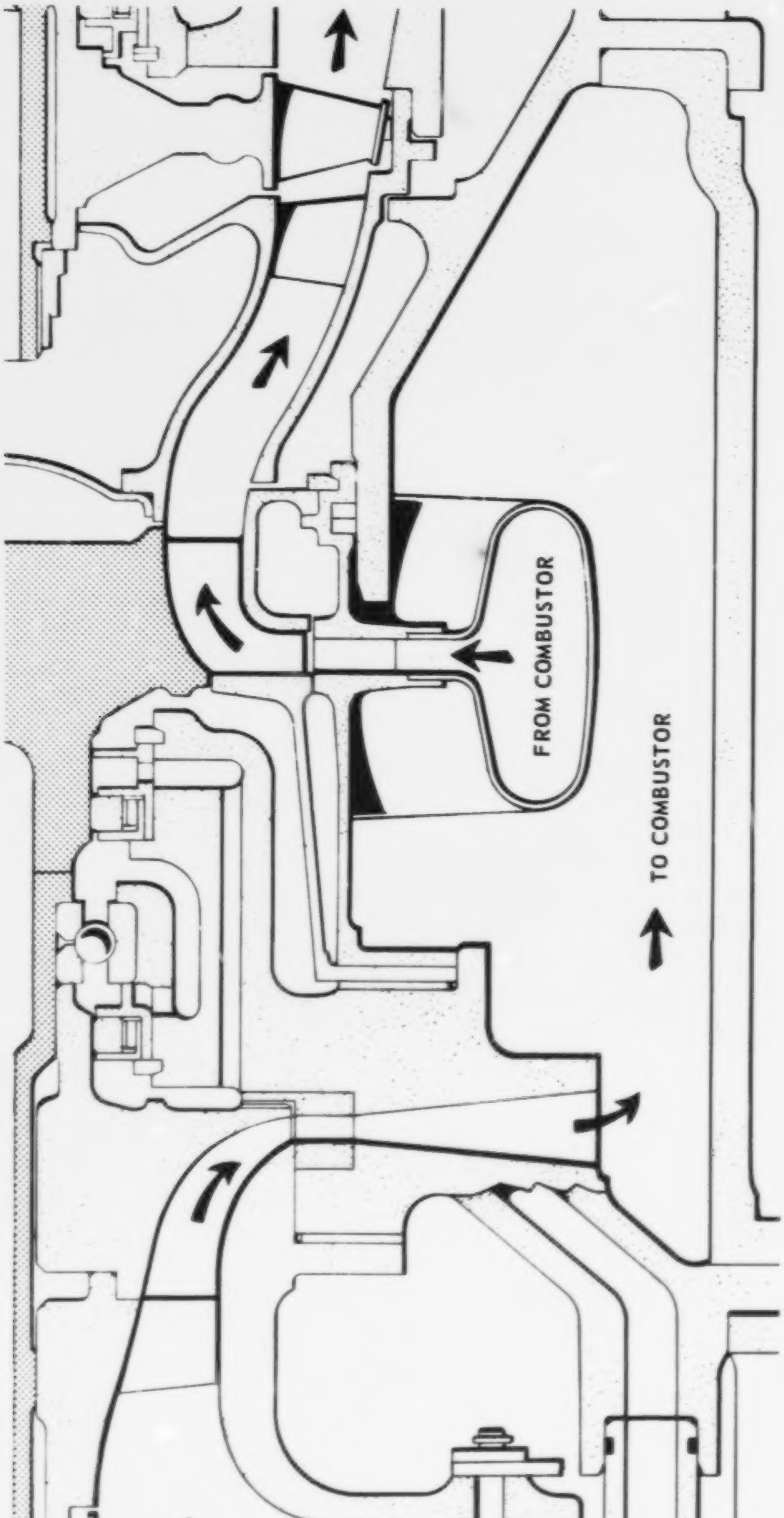
FIG. 11

K-971017-4


FROM COMBUSTOR
TO COMBUSTOR

SKETCH OF GAS PATH, FULL SIZE, SFT-12

The performance of the SFT-12 engine, which also has a nominal output of 150 hp, is presented in Fig. 12 for standard-day conditions. The performance shown corresponds to the use of an optimum schedule of inlet guide vane settings, and includes the optimization and installation loss effects noted for Fig. 9.

RSS-7

The layout of the advanced regenerated engine is shown in Fig. 13. This engine incorporates a rotary drum-type regenerator in which the flow is radial and at right angles to the axis of rotation. The engine is of single-shaft configuration, and output for both power and accessories is to the front (i.e., cold) end of the engine. The radial air intake located approximately at the center of the engine permits air to pass through a set of fixed guide vanes to the centrifugal impeller.

The centrifugal impeller is of a design developed at UACL, and its predicted performance is shown in Fig. 14. This compressor is, again, the key component which distinguishes this engine from current low-pressure-ratio regenerative gas turbine engines. The compressor map is based on compressor characteristics measured at a size and at speeds close to those being considered, and at actual airflows not exceeding 1.25 lb/sec. A 2% increment of efficiency has been added to measured efficiencies to account for 1976 levels of expected capability. Test compressor measurements include inlet losses and are obtained with realistically achievable running clearances of about 0.020 inches. The diffuser is the patented UACL pipe diffuser.

The impeller is driven by a radial turbine which exhausts to the regenerator through a set of turning vanes. A single-can combustor feeds the combustion products into an annular scroll for expansion through the turbine nozzle ring. The shaft speed is 112,000 rpm at the design point, which is reduced by planetary reduction gearing to 12,555 output rpm. As is the case for the SSS-12 engine, an actual production engine would be designed for a lower output speed (in the 3600 to 10,000 rpm range) to be compatible with probable transmission configurations. Also included on the output pad are drives for belts, a fuel control, and a starter-generator.

The regenerator shown includes 20 stainless steel wire-screen heat exchanger matrix packs assembled in a cast-iron carrier with a rectangular seal assembly. The device rotates through two cold air tunnels and is driven by a separate hydraulic motor at 22 rpm. Although it is shown with the RSS-7, there is nothing inherent in the design to preclude its being applied to the RFT-4 should it prove feasible. Conversely, there is nothing to prevent the ceramic disk regenerator of the type shown for the latter from being applied to the advanced engine. (In fact, to do so would entail a lower technical risk for 1976-78 production requirements, and might therefore be the preferred approach.) This advanced regenerated

FIG. 12

ENGINE PERFORMANCE ESTIMATES, SFT-12

COMPOSITE OF OPTIMUM INLET GUIDE VANE SETTINGS
5% INSTALLATION LOSSES INCLUDED

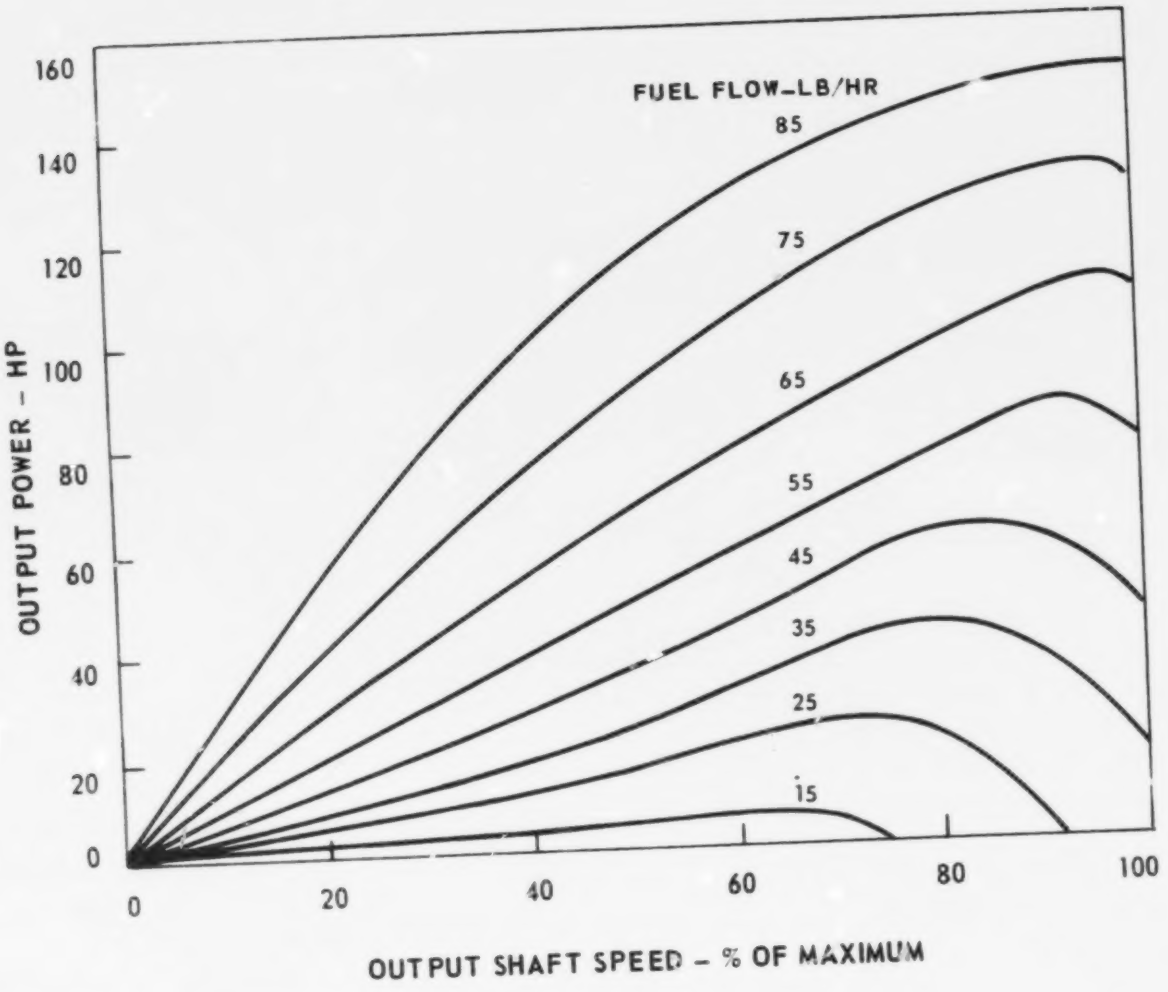
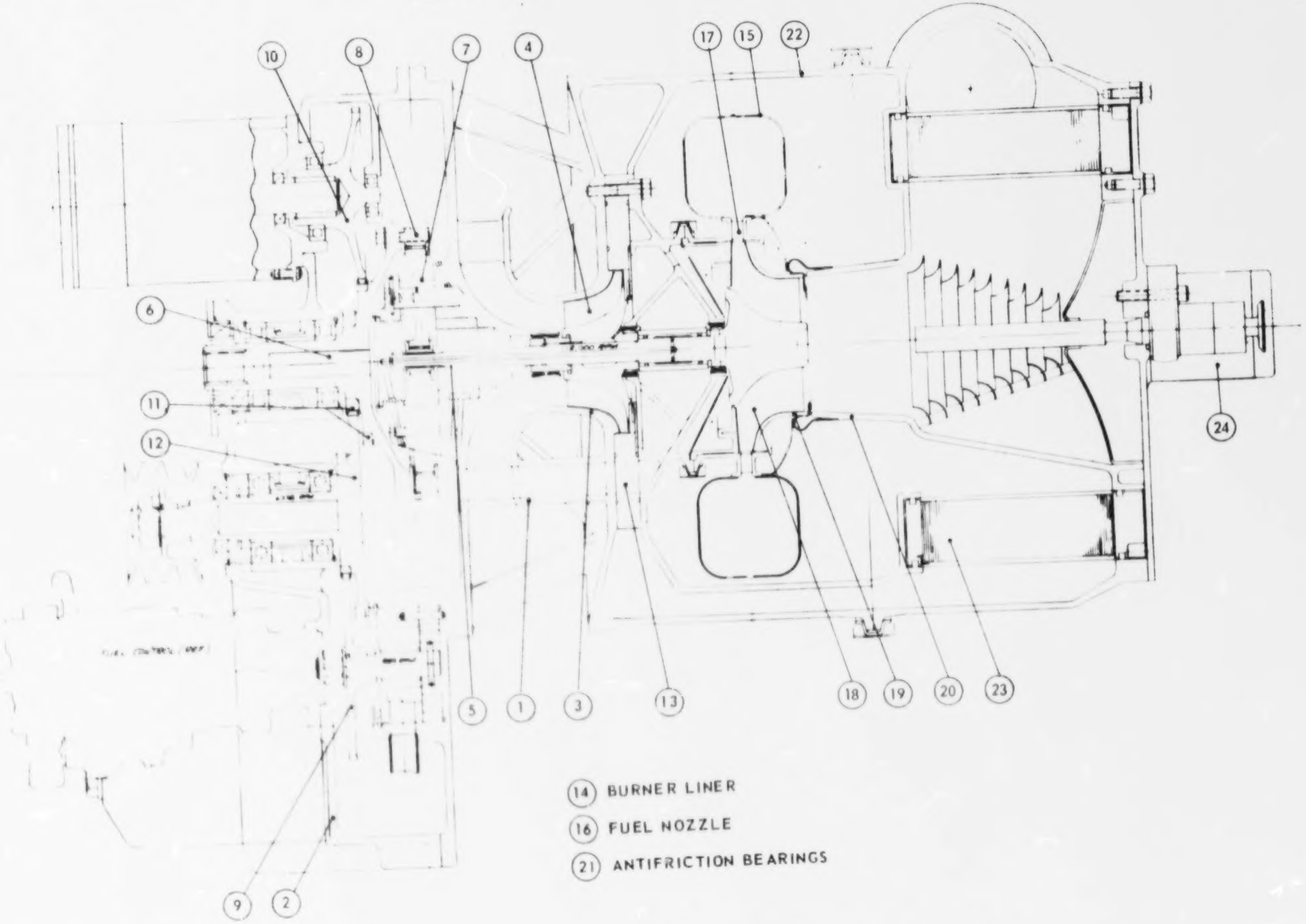


FIG. 13

LAYOUT, REGENERATED SINGLE-SHAFT 7:1 ENGINE (RSS-7)

NOTE: PART NUMBERS ARE KEYED TO TABLE XII


10
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11
12
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3
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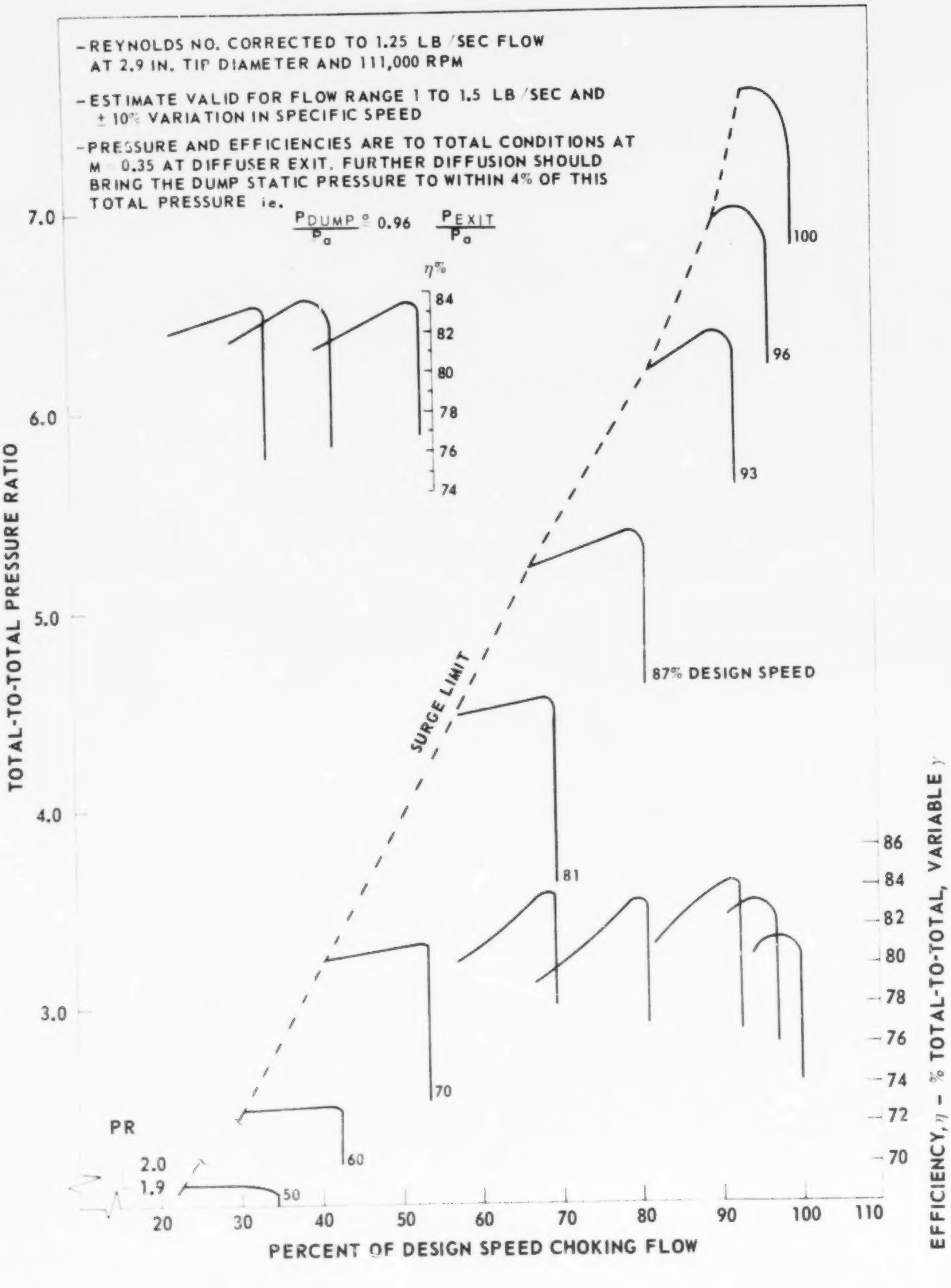
23
9
2
(14) BURNER LINER
(16) FUEL NOZZLE
(21) ANTIFRICTION BEARINGS

FIG. 14

K-971017-4

COMPRESSOR PERFORMANCE ESTIMATES, RSS-7



- REYNOLDS NO. CORRECTED TO 1.25 LB/SEC FLOW AT 2.9 IN. TIP DIAMETER AND 111,000 RPM
- ESTIMATE VALID FOR FLOW RANGE 1 TO 1.5 LB/SEC AND ± 10% VARIATION IN SPECIFIC SPEED
- PRESSURE AND EFFICIENCIES ARE TO TOTAL CONDITIONS AT M = 0.35 AT DIFFUSER EXIT. FURTHER DIFFUSION SHOULD BRING THE DUMP STATIC PRESSURE TO WITHIN 4% OF THIS TOTAL PRESSURE ie.

$$\frac{P_{DUMP}}{P_a} \approx 0.96 \frac{P_{EXIT}}{P_a}$$

engine weighs 300 lb, which is somewhat less than it would weigh if designed with a ceramic-matrix regenerator.

The performance of the advanced regenerated engine* is shown in Fig. 15. It will be noted that the torque curve drops sharply with decreasing speed (which is characteristic of all single-shaft gas turbine engines), so that it requires a more complicated transmission than that for the free-turbine engine shown in Fig. 12, the incremental cost of which has to be traded against the additional cost of a free turbine. In considering this tradeoff, it should also be noted that the best operating line (with regard to fuel efficiency) is quite close for both engines. Therefore, where efficiency is a problem, as has been the case in automobile gas turbines to date, both engines might well require a more complicated transmission, so that even in the case of the free-turbine engine, the more complicated transmission might be chosen in order not to impose too severe a penalty in fuel economy.

RFT-4

A layout of the low-pressure regenerative engine which was designed by H. J. Wood and Associates to be representative of conventional automobile gas turbines is shown in Fig. 16. It is a 4:1-pressure-ratio engine with a rotary regenerator composed of a single ceramic disk of Cercor material driven by a hydraulic motor and located in the top plane of the engine. The single ceramic disk regenerator configuration was chosen to minimize the cost of seals and disks.

The turbomachinery consists of a gas generator shaft with a centrifugal compressor driven by a radial turbine followed by an axial single-stage power turbine preceded by variable turbine nozzle vanes. The compressor map upon which the performance estimates are based is shown in Fig. 17. It is believed to be fairly representative of current technology but has a 2% improved efficiency predicted to be achievable after a 5-year development program. A conventional vane-type diffuser is used. The combustor is a single-can unit, located off-center from the engine centerline, which feeds combustion products into an annular scroll for distribution to the radial turbine nozzle vanes.

The front of the engine houses the air intake for the compressor and a power takeoff for accessories which include a belt-drive pulley, a starter pad, and a fuel control pad. The actual power output of the engine for input to the transmission is at the rear (i.e., hot) end of the engine. The output is at 6000 rpm from a two-stage reduction gear. Rotor speeds within the engine at the design point are 66,500 rpm for the gas generator shaft, and 71,900 rpm for the power turbine.

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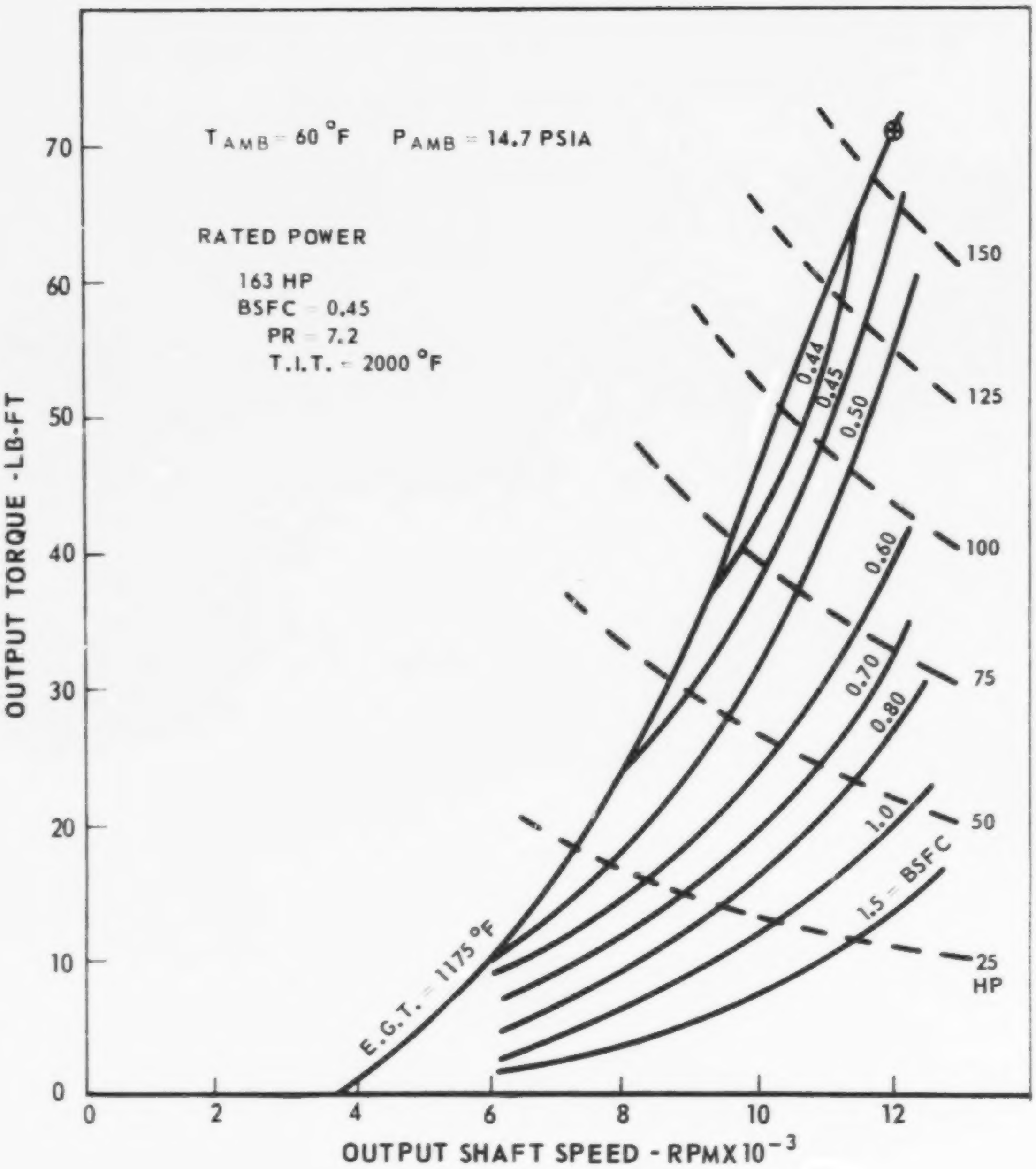
* Estimates for the entire performance range of the regenerated engines were prepared by the design subcontractor, H. J. Wood and Associates.

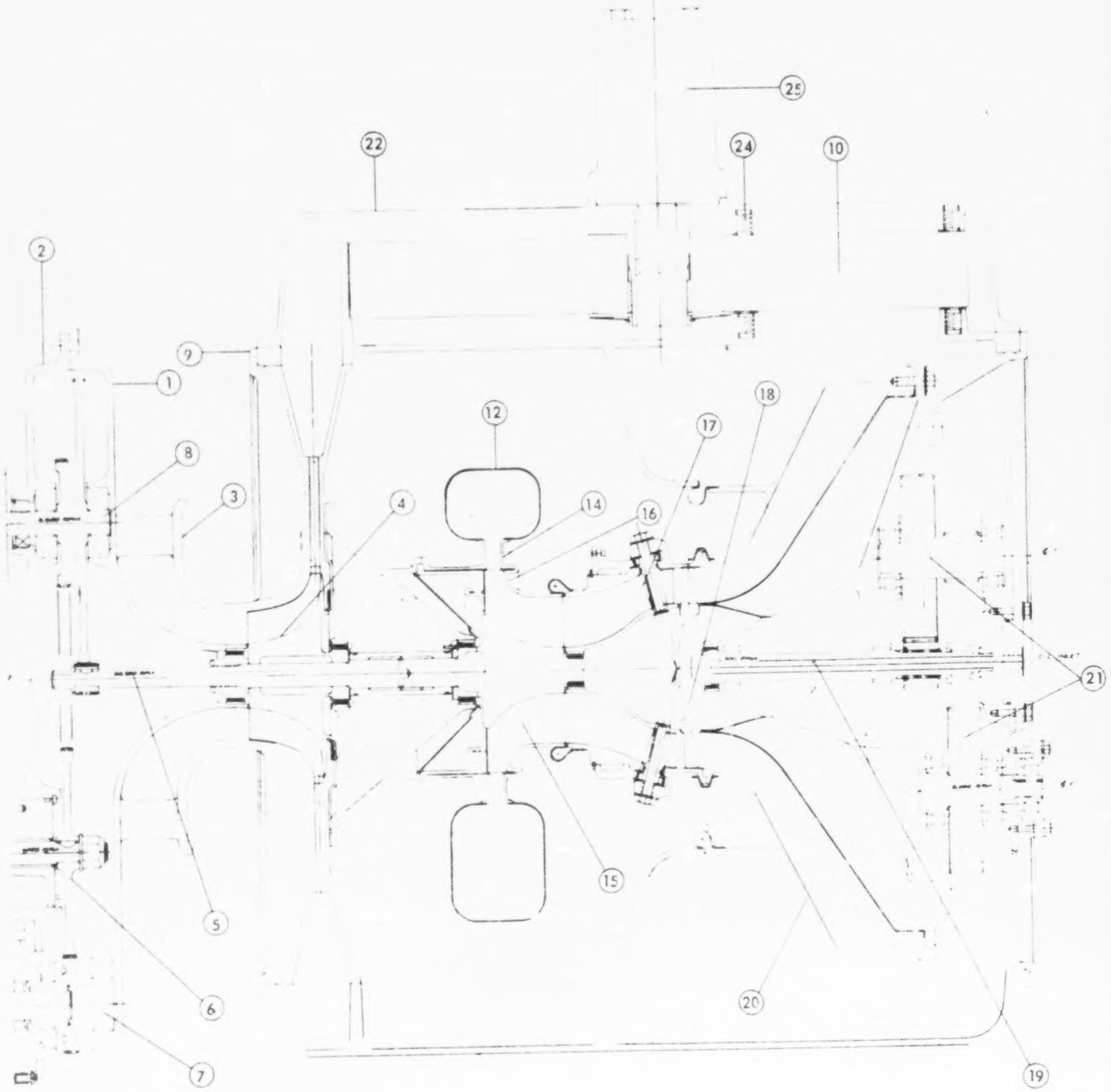
FIG. 15

ENGINE PERFORMANCE ESTIMATES, RSS-7



LAYOUT, REGENERATED FREE TURBINE 4:1 ENGINE (RFT-4)

NOTES: PART NUMBERS KEYED TO TABLE XIII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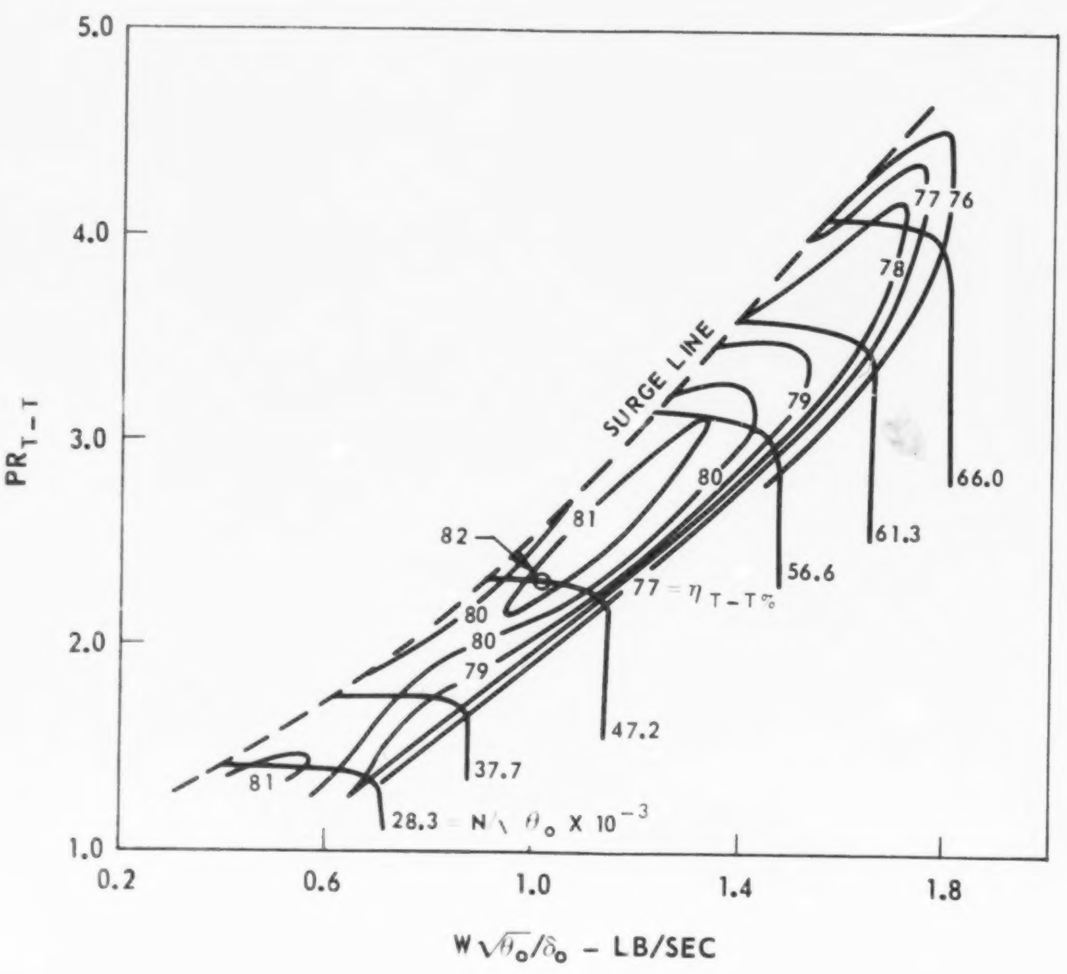
(11) BURNER LINER

(13) FUEL NOZZLE

(23) BEARINGS

COMPRESSOR PERFORMANCE ESTIMATE
RFT-4



The performance of the basic engine is described in Fig. 18 for standard-day conditions. The engine exhibits the characteristic torque curve of free-turbine engines, i.e., torque increases linearly as output speed is reduced, and the maximum stall torque is approximately twice the rated torque at the design point. Operation of the engine at maximum torque at low speeds entails a considerable fuel cost penalty, and it will be noted that the optimum fuel consumption speed for the regenerated engine closely follows the shape of that expected for the single-shaft regenerated engine previously discussed. The design point characteristics of this engine are also given in Table I (Page 16).

Size and Weight Comparisons

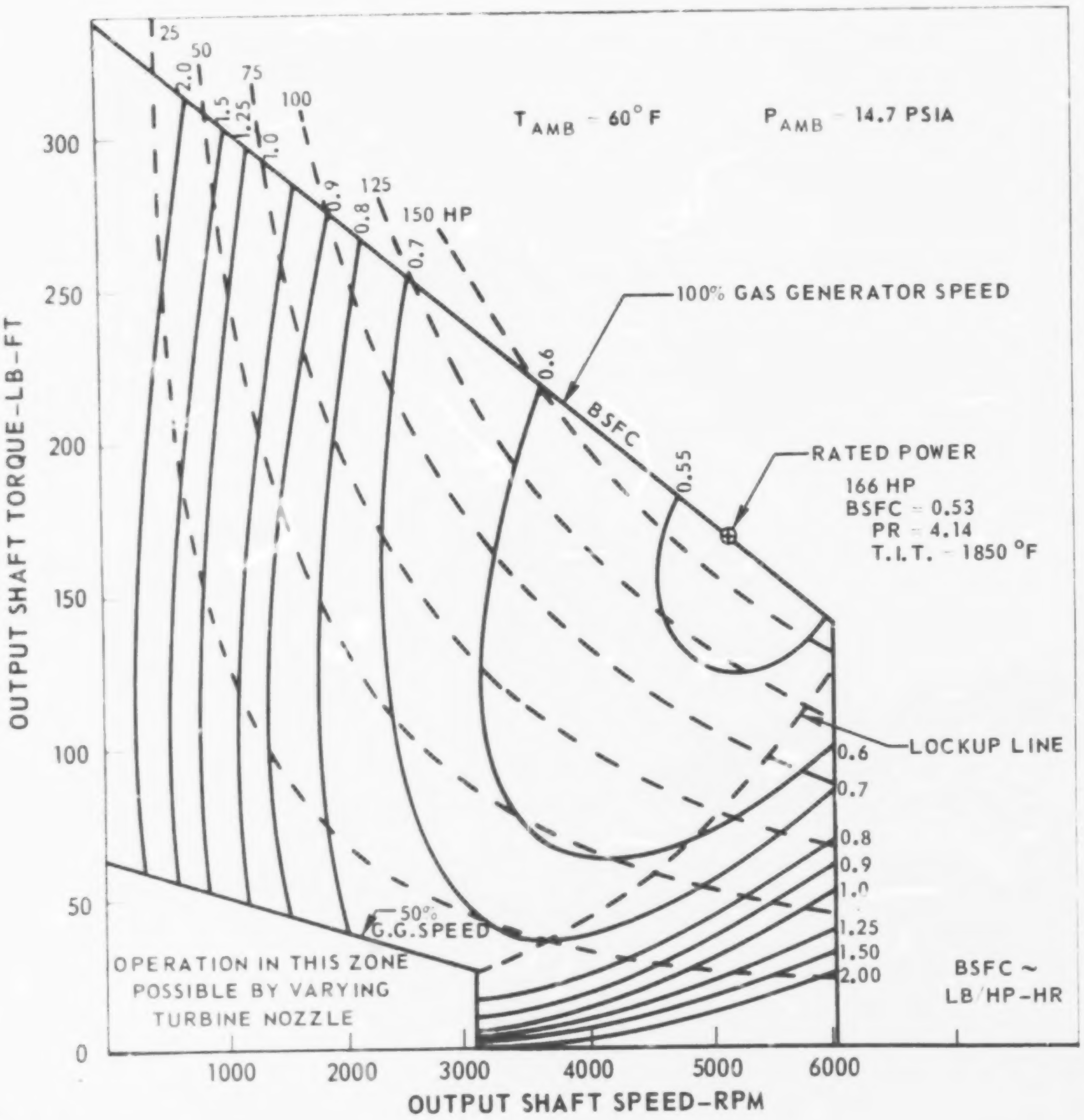
The comparable sizes of the study engines are illustrated in Fig. 19 which shows sketches of the four gas turbine engines approximately to scale. Since the space required for installation of the different engines depends strongly on the details of the actual installation itself, which can vary widely from vehicle to vehicle, a wholly significant size comparison cannot be made. In general, ducting for intakes and exhausts of automobile gas turbines will have greater volumetric requirements than those for corresponding piston engines since their mass flows are larger, their exhaust temperatures are higher, and they are more sensitive to intake and exhaust pressure drops. Some of the apparent required installation volume advantage of the gas turbine engines may thus be lost in an actual installation.

Weight comparisons among the engines considered (as well as estimates for comparable 1970 and 1976 Otto-cycle engines) are shown in Table II. The first line represents the weight estimated for the engine alone (without accessories), while line W_1 represents the more significant weight of the complete engine, including accessories. Line W_2 shows estimates for those elements of the propulsion system other than the engine (including the weight of a full tank of fuel), while line W_3 (W_p) shows the corresponding weights of the entire propulsion system. It will be noted that both simple-cycle engines (SSS-12 and SFT-12) have a propulsion system weight estimated to be approximately 450 lb (35%) less than the standard present installation, while the corresponding figures for the RSS-7 and RFT-4 engines are approximately 400 lb (30%) and 225 lb (17%) less, respectively. These weight savings are shown to be significant in both the acceleration analysis and ownership cost analysis discussed in later sections of this report.

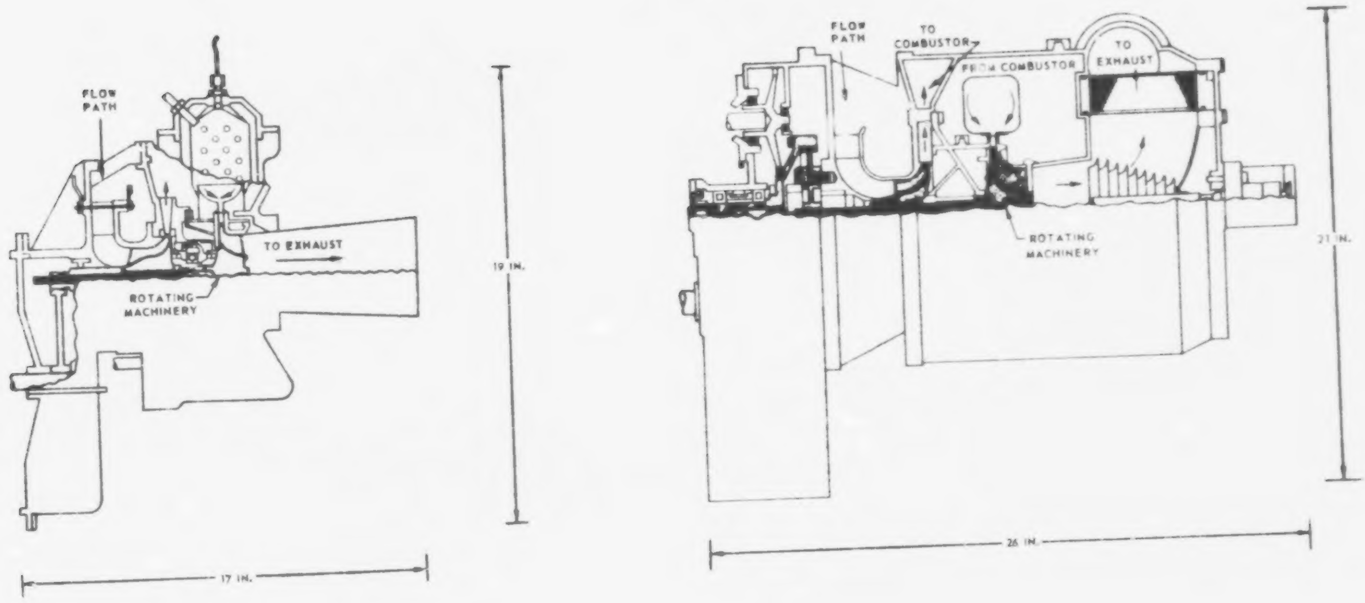
Negative Torque

The negative torque developed by the engine at idle fuel flow is important in determining the amount of braking available from the engine. The negative torque is a function of the engine configuration (single-shaft or free-turbine), the compressor characteristics, the idle fuel flow (or deceleration fuel flow), and the

ENGINE PERFORMANCE ESTIMATE, RFT-4



SUMMARY OF ENGINE DESIGNS APPROXIMATELY TO SCALE

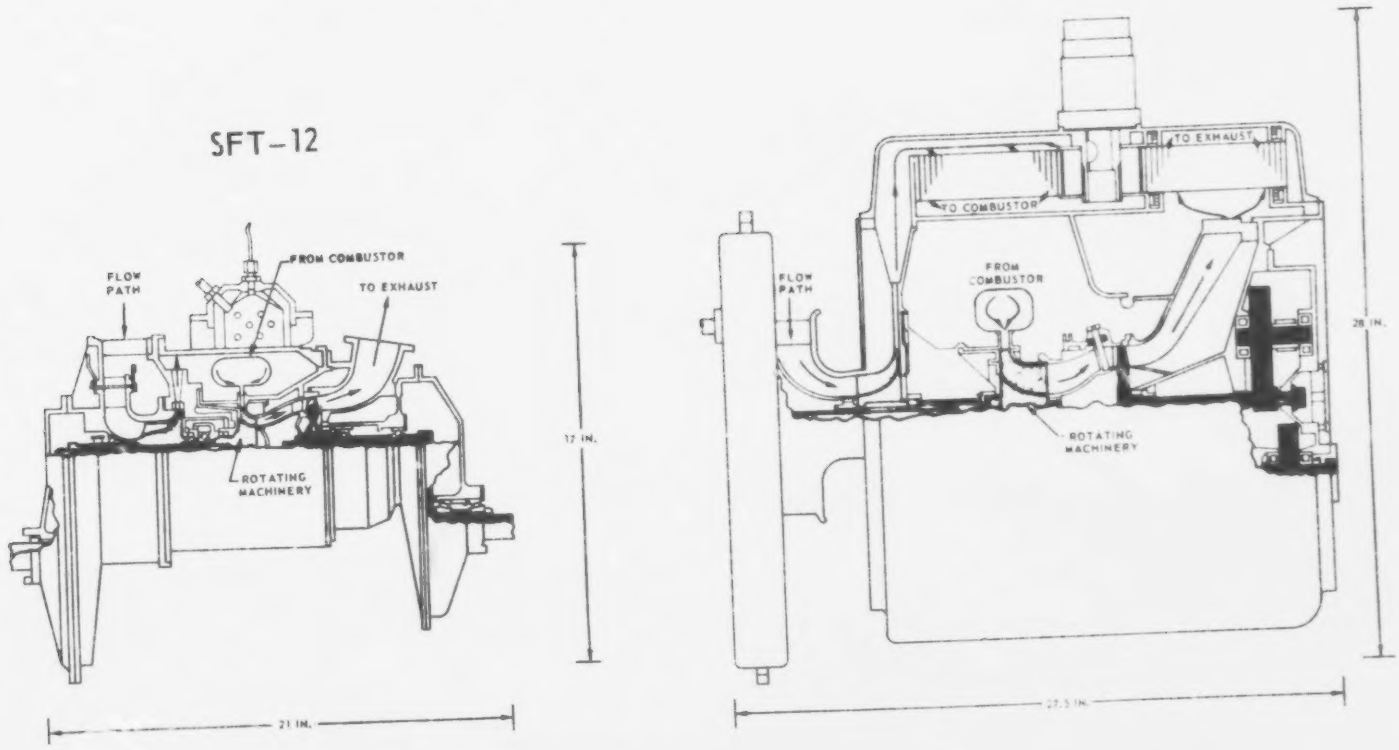


SSS-12

RSS-7

FLOW PATH
TO EXHAUST
ROTATING MACHINERY
19 IN.
17 IN.
FLOW PATH
TO COMBUSTOR
FROM COMBUSTOR
TO EXHAUST
ROTATING MACHINERY
21 IN.
26 IN.

RFT-4



SFT-12

FLOW PATH
FROM COMBUSTOR
TO EXHAUST
ROTATING MACHINERY
17 IN.
21 IN.
TO EXHAUST
TO COMBUSTOR
FLOW PATH
FROM COMBUSTOR
ROTATING MACHINERY
28 IN.
27.5 IN.

TABLE II

ESTIMATED VEHICLE WEIGHTS

	Computation	Basic* Vehicle %	Weight - lb					
			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Engine		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Radiator		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Starter		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Generator		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
W_1 - Engine with Accessories		17.5	700	740	160	200	290	490
Transmission		4.0	160	[illegible]	185	135	150	135
Rear Axle and Drive Line		4.4	175	170	175	175	170	180
Battery		1.1	45	45	45	45	45	45
Fuel and Tank (200 mile range)		4.4	175	170	215	245	140	160
Exhaust		1.3	50	100	75	75	75	75
W_2 - Propulsion Accessories		15.2	600	670	690	640	615	585
W_3 (W_p) - Propulsion System	$W_1 + W_2$	32.7	1300	1410	850	840	905	1075
Suspension		6.0	240					
Wheels		2.5	100					
Tires		3.3	130					
Brakes		3.9	155					
Steering		1.6	65					
Part of Body (Chassis-Frame)		11.0 **	440					
W_4 - Nonpropulsive Variable Items		28.3	1130					
Remainder of Body		22.0	880					
Trim		14.0	560					
Glass		3.2	130					
W_5 - Nonpropulsive Fixed Items		39.2	1570	1570	1570	1570	1570	1570
W_6 (W_v) - Vehicle Without Propulsion System	$W_4 + W_5$	67.5	2700	2700	2700	2700	2700	2700
W_7 (W_c) - Vehicle Curb Weight	$W_3 + W_6$	100.0	4000	4110	3550	3540	3605	3775
W_8 (W_t) - Vehicle Test Weight	$W_7 + 300$	--	4300	4410	3850	3840	3905	4075
W_9 (W_g) - Vehicle Gross Weight	$W_7 + 1000$	--	5000	5110	4550	4540	4605	4775
W_4* - Adjusted W_4	$0.292\ W_3 + 750$		1130	1160	1000	1000	1015	1065
W_6* - Adjusted Vehicle without Propulsion System	$W_4* + W_5$		2700	2730	2570	2570	2585	2635
W_7* - Adjusted Curb Weight	$W_3 + W_6*$		4000	4140	3420	3410	3490	3710
ΔW_7* - Referred to 1970 Otto Cycle			0	+140	-580	-590	-510	-290
W_8* - Adjusted Test Weight	$W_7* + 300$		4300	4440	3720	3710	3790	4010
W_9* - Adjusted Gross Weight	$W_7* + 1000$		5000	5140	4420	4410	4490	4710

* Hoffman (Ref. 12)

** UARL assumption of portion of body affected by propulsion system weight.

use or lack of use of special engine braking devices such as variable-geometry power turbine nozzles or power transfer clutches. In addition, the actual braking effect which the engine can deliver to the wheels is strongly affected by the transmission. Figure 20 displays estimated typical negative torque curves for the gas turbine engines studied. The maximum negative torque is generated by the single-shaft engines (SSS-12 and RSS-7), and is the direct result of coupling the output shaft directly to the compressor so that the compressor acts as an energy-absorbing pump. Between these two engines, the difference in the magnitude of the negative torque at any speed is a direct result of the lower idle fuel flow selected for the regenerated engine; the negative torque could be equalized (or otherwise adjusted) by appropriate adjustments of the idle (or deceleration) fuel flow. The difference in the range (% output speed) over which braking is available from the single-shaft engines is due to the differing characteristics of the compressors in these engine designs.

In the case of the free-turbine engines, the output shaft is not coupled mechanically to the compressor shaft; in this case, the unassisted degree of braking available from the free turbine is displayed by the SFT-12 curve. Because of its lack of engine braking, this engine may be unsatisfactory for an automobile engine unless sufficient braking action could be provided in the transmission through the use of a hydraulic retarder or similar device. Since this procedure may not be acceptable from a safety standpoint, a modification to the design of the engine may be required, such as including variable-geometry power turbine nozzles, or power transfer clutches. The RFT-4 negative torque characteristic is achieved by scheduling its variable-geometry power turbine nozzles to alter gas flow to provide negative torque from the power turbine.

Fuel Controls

The cost of the fuel controls is obviously important in evaluating the feasibility of manufacturing engines at a competitive cost for the automobile application. However, the limited scope of the study prevented the treatment of fuel control systems in as much detail as that applied to the basic engines discussed above. Furthermore, firm requirements for the fuel control are not available at this time, since such requirements depend heavily on customer reactions in demonstrator vehicles, and additional features may be required to meet emissions requirements. Nevertheless, existing fuel control systems for automotive use for both gas turbine and reciprocating engines were evaluated, and the modifications necessary to adapt them to the study engines were considered.

Discussions concerning fuel controls for the engines considered in this report were held with the Hamilton Standard Division of United Aircraft and the Holley Carburetor Division of Colt Industries*. An existing Holley R550 control is shown

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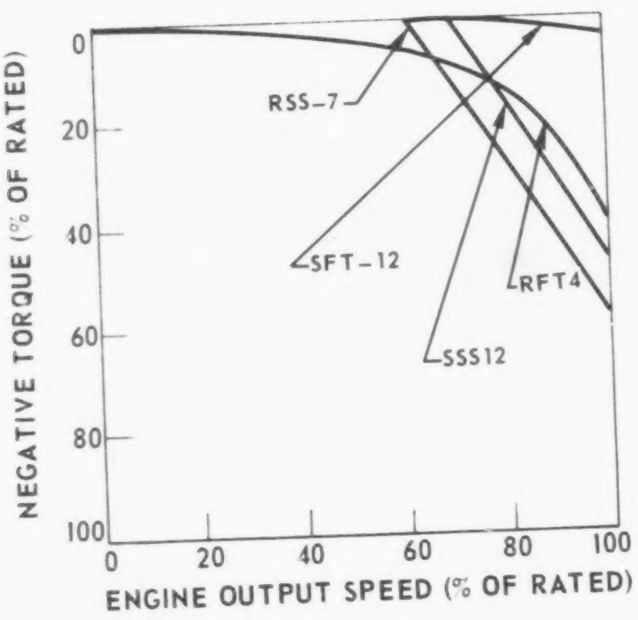
* Turbine Control Systems

FIG. 20

TYPICAL NEGATIVE TORQUE CURVES



in Fig. 21. While this control was not designed directly for this application, and is consequently oversized in fuel flow, it nevertheless is typical of a possible hydromechanical approach to the problem.

The basic hardware of the Holley R550 Fuel Control is intended to serve the general gas turbine market in a maximum fuel flow range of 200 to 1000 lb/hr. The R550 has its own pump and provides starting and acceleration flow metering, all speed droop governing, a minimum-fuel flow system, and manual fuel shutoff. These features are considered basic functions and are shown in Fig. 21.

The fuel pump is a nested version of a conventional spur gear-type unit. To insure long life, a filter is required upstream of the pump inlet. This filter will handle metered flow quantities only, since excess pump flow is bypassed internally to the pump inlet.

The spring determines engine flow during starting when the engine compressor discharge pressure, CDP, is negligible. As the engine speed increases after light-off, CDP rises. The pressure sensed by the control, CDP', is a function of CDP determined by the size and characteristics of an inlet and exit restriction. A great deal of flexibility can be obtained by proper design of these restrictions.

The deceleration fuel flow is a fixed percentage of the acceleration fuel schedule. This deceleration fuel schedule is controlled by the minimum stop on the fuel valve and the acceleration fuel system as applied to the bypass valve.

As the engine speed approaches that demanded by the throttle position, the speed-sensing pressure signal, SS-BG, which is generated by a rotating throttling valve which balances SS-BG against centrifugal forces, creates a force which gradually overcomes that of the governor spring. This causes the governor valve to move in a closing direction, reducing flow to the engine until it stops accelerating.

If the throttle is moved to call for reduced speed, the governor valve can reduce flow to the engine until prevented from doing so by the minimum-flow stop, or by the valve seating. In the latter case no fuel would be sent to the engine.

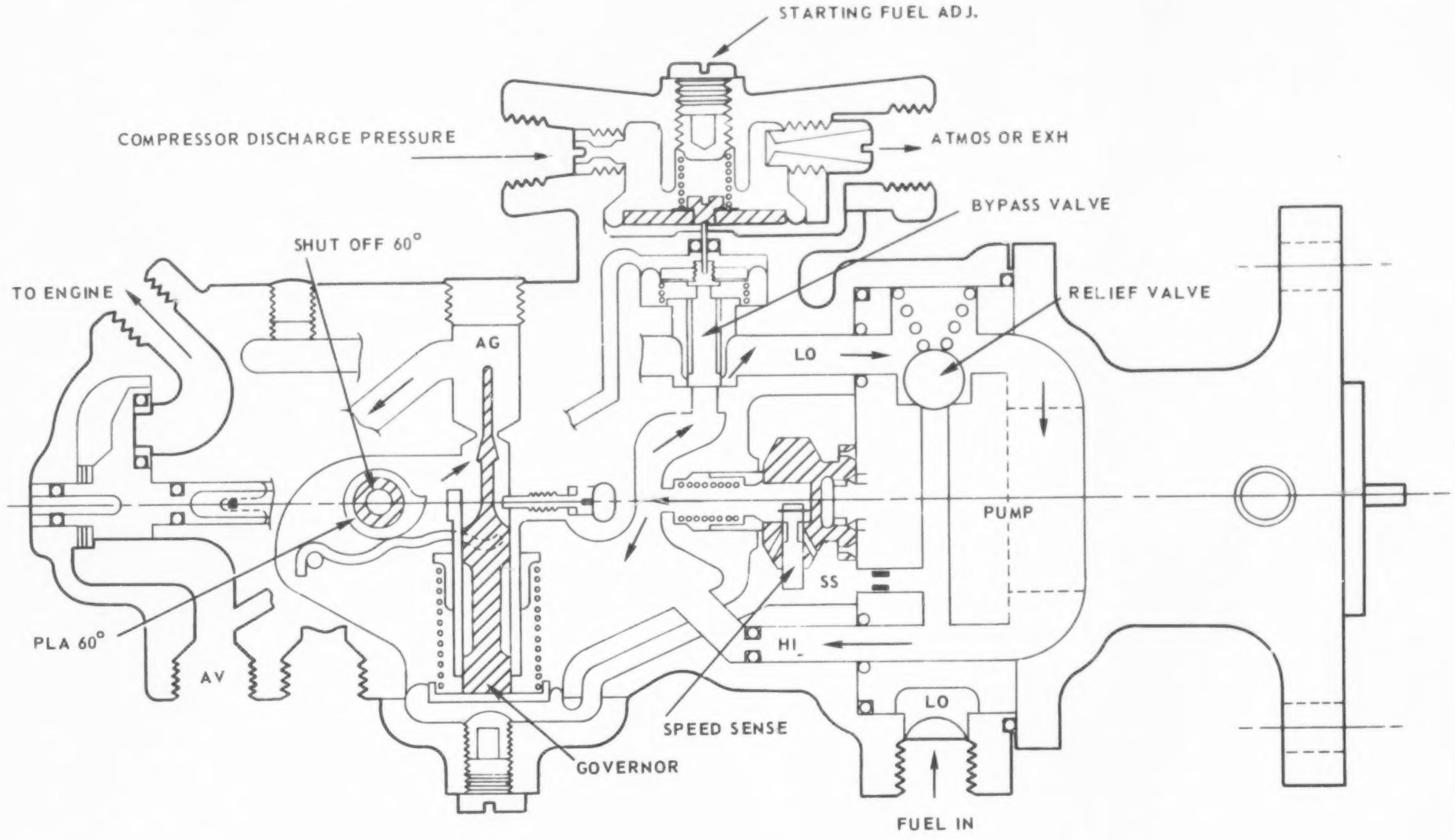
The cost of any fuel control optimized for the low cost necessary for the automobile application will depend primarily on the number and type of functions it must perform. Simpler controls (with fewer functions than the one shown in Fig. 21) could possibly be designed to operate effectively in the standard automobile applications.

As mentioned above, functional requirements for the basic fuel control can be determined only after extensive testing in the field, ultimately including determination of customer reactions. However, as in the case of present automobile

FIG. 21

TYPICAL HYDROMECHANICAL FUEL CONTROL

(HOLLEY R550)


STARTING FUEL ADJ.
COMPRESSOR DISCHARGE PRESSURE
ATMOS OR EXH
BYPASS VALVE
SHUT OFF 60°
TO ENGINE
RELIEF VALVE
AG
LO
PUMP
SS
HI
PLA 60°
AV
LO
SPEED SENSE
GOVERNOR
FUEL IN

engines, the best approach to the problem is probably to design the engine itself somewhat oversized (for the basic performance required) with a relatively cheap, low-performance control, and to provide the customer with performance options at additional cost, - just as several fuel-control-system options are commonly offered with automobiles now, such as performance options involving fuel-control modifications (i.e., four-barrelled carburetors), improved intake manifolds, and possibly different cams.

Power Transmission

The engine itself obviously represents only one component of the power transmission system, and the output characteristics of the engine must be modified by a transmission in order to provide suitable traction power. Torque-speed characteristics and estimated weights of both single-shaft and free-turbine engines were provided to Borg-Warner* with the request that they recommend suitable transmission designs and evaluate these designs in their vehicle performance computer programs.

The transmission recommended by Borg-Warner for the free-turbine engine is a four-speed, completely automatic gearbox without the customary torque converter used in present engines. Transmissions of this type are expected to be in production shortly, and with the elimination of the torque converter, they can be expected to cost somewhat less and be somewhat lighter than conventional automatic transmissions for present-day automobiles.

The single-shaft engine represents a major transmission problem, because the rapid decrease of available torque as engine speed declines requires sophisticated transmission output characteristics, approaching those of infinitely variable transmissions, in order to provide power characteristics comparable with those of current automobiles. Consequently, for the single-shaft engine, an eight-speed transmission coupled to a controlled-speed slipping clutch was recommended by Borg-Warner. The eight-speed automatic transmission consists of a four-speed automatic gearbox preceded by a two-stage planetary splitter which, in effect, doubles the number of ratios attainable from the gearbox itself. The controlled slipping clutch transmits constant torque to the wheels at any engine speed, without multiplication. Its advantage is that the stall speed can be adjusted in the design process so that the engine can be completely idle or be completely engaged within a fairly narrow range. In addition, when the clutch is locked up, power is transmitted with a higher efficiency than that associated with the torque-converter device.

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* The Borg-Warner Research Laboratories furnished consultation in the area of drive-line transmissions, and this section is based largely on their work.

The performance predicted by Borg-Warner's computer simulations is shown in Fig. 22. Both gas turbine engines provide somewhat better acceleration performance over the majority of the speed range than does the conventional Otto-cycle engine with its conventional transmission. The improvement is primarily due to the reduced weight of the gas turbine-powered vehicles, which is attributable to the lighter weight of their engines. Nevertheless, such performance cannot be obtained without highly satisfactory transmissions.

The controlled slipping clutch is not new to automotive applications. Its use was demonstrated by Borg-Warner during automotive transmission programs of the mid-1950's, and several prototype vehicles using such a clutch were built. More recently it has been studied extensively as a device suitable for coupling unconventional internal combustion engines, such as the Wankel engine, with gearboxes suitable for automobile production. It should be noted that controlled slipping devices have also been demonstrated extensively by General Motors in the GT309 vehicular gas turbine engine, in which a power transfer device is used to transfer scheduled amounts of power between the free turbine and the gas generator shaft for purposes of engine braking and improved part-load fuel economy. Power losses and wear rates with such devices appear to be reasonable for the relatively short life and low average power requirement of automobiles.

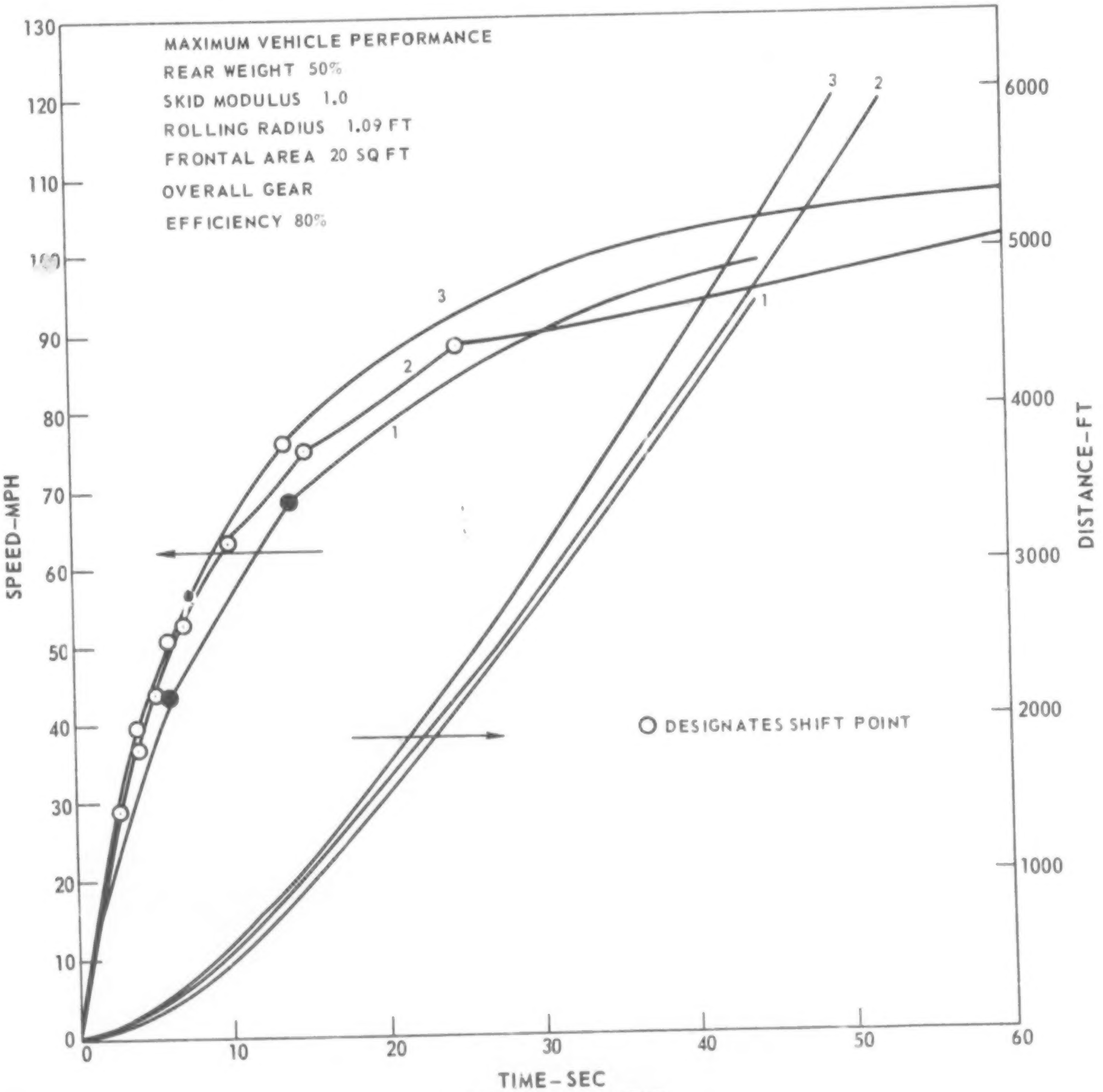
With regard to the manufacturing cost comparisons of the various transmissions considered in this report, no definitive results can be given at present. An extensive design study, layout, and costing exercise of the suggested transmissions is required to provide accurate estimates. It has been stated by Borg-Warner that the increment in direct manufacturing cost (to the automobile manufacturer) which might be incurred by the transmission for the single-shaft engine might be from $0 to $50 greater than a conventional automatic transmission. On the other hand, the cost of the transmission for the free-turbine engine should be less than that presently used with the Otto-cycle since the torque converter is eliminated. Appropriate allowances for these considerations are made in the ownership cost analysis in the concluding section of this report.

Although comprehensive surveys of transmission manufacturers were not conducted as part of this study, it is known that other apparently feasible approaches exist for transmissions suitable for single-shaft engines. One of these is under development by the Ordnance Division of the General Electric Corporation and is known as the HMT (HydroMechanical Transmission, Ref. 10). It is a split-path, hydrostatic-planetary device which is infinitely variable. Although its efficiency is lower than that of the 8-speed transmission discussed above, these losses are partially compensated, in terms of fuel economy, by permitting the operation of the engine along its optimum fuel economy line, and more than compensated, in acceleration comparisons, by permitting full engine power to be delivered to the transmission input at all nontraction-limited vehicle speeds. It is the manufacturer's contention that no significant production cost penalty is involved compared

FIG. 22

COMPARISON OF TRANSMISSION PERFORMANCE

CURVE	1	2	3
ENGINE TYPE VEHICLE WEIGHT	OTTO CYCLE 3850 LBS	SINGLE SHAFT 3500 LBS	FREE TURBINE 3500 LBS
AXLE RATIOS	3.25	7.56*	7.20*
GEAR RATIOS	2.4, 1.46, 1.0	4.5, 3.3, 2.56 2.06, 1.69, 1.40 1.18, 1.00	3.0, 2.2, 1.45 1.0
ENGINE TO TRANSMISSION COUPLING	CONVERTER	CONTROLLED SLIPPING CLUTCH	DIRECT



*FOR 10,000 RPM INPUT

with present transmissions. Estimated performance comparisons using the HMT transmission are presented elsewhere in this report.

Accessories and Installation

A very wide series of problems falls under the heading of accessories, installation, systems integration, maintenance, etc. In all these cases, the approach used was to identify the problems (if any) and to determine if there appeared to be practical solutions to the problems. Solutions were found in all cases, but no further optimization was performed.

Insofar as accessories are concerned, it was determined that there may be no appreciable differences between the alternator and starter requirements of gas turbines and those associated with conventional automobile engines which would affect the costs of these components. It is probably possible to design a single-unit starter-generator for the gas turbine engine, which might result in some production cost savings over existing configurations. On the other hand, the relatively slow startup times of the gas turbines (up to 30 seconds) may be undesirable to the customer, and the installation of more powerful (and costly) starters and batteries may be required. The oil flow requirements of the engine are quite similar to present-day automotive engines, and oil pump prices were established accordingly. Inlet air requirements were not treated in detail; a paper filter inlet air cleaner is visualized as being satisfactory for all engines considered, which would be similar in function, but somewhat larger, than those presently installed in automotive engines. Exhaust installations were not considered in detail, although they represent a definite design problem due to increased mass flow and relatively high tailpipe temperatures at idle.

Whereas a significant advantage would accrue to the gas turbine (at least the high-pressure-ratio gas turbine) with the use of direct-cycle air conditioning systems, the vehicles considered herein were not assumed to be necessarily equipped with air conditioning. Therefore, this aspect was not investigated in detail. Nor was location of the engine within the vehicle investigated in any detail. It was assumed that the conventional engine location (at the front of the car) with a direct connection to a suitable transmission would be acceptable. Because of the small size and weight of the simple-cycle engine, it is possible that more advantageous locations for the power system could be found (mid or rear) or that unconventional orientation (transverse or vertical) would prove desirable.

Maintenance considerations have not been treated extensively in this study, primarily because of the lack of applicable field experience, although some cursory estimates for ownership costs are discussed on page 147. It is expected that the maintenance requirements for the simple-cycle single-shaft engine will be minimal primarily because of its basic simplicity, and its servicing requirements are expected to be limited mainly to the air, oil, and fuel filters. Access to the fuel, combustion, and ignition systems will be possible in the field, but these service requirements should be minimal. Since the engine is so compact, and since training and equipping existing service centers may be difficult or uneconomic, the remove-and-replace concept of the basic engine appears attractive, with overhaul performed to the extent necessary by centrally located depots. In the case of the regenerated engines, maintenance requirements are probably increased due to the additional complexity of the engine. However, manufacturers of cores and seals appear to be confident that the problems associated with these components will be solved by 1975.

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AUTOMOBILE PERFORMANCE ESTIMATES

General

One of the tasks performed under this contract involved the development and exercise of an automobile performance computer program which would determine the power requirements, fuel consumption, and emissions for a given vehicle and duty cycle, in order to obtain comparative data on the gas turbine engines considered in the study and an Otto-cycle engine typical of conventional automobile power plants.

The elements of this task may be grouped as follows:

1. Development of the automobile performance computer program
2. Generation of fuel consumption and emissions data for the four types of engines considered in this study
3. Selection and definition of automatic transmissions to which the study engines could be matched
4. Definition of a base-line vehicle, in terms of weight, drag coefficient, etc., compatible with the study engines and transmissions, as well as with certain Federal vehicle design goals
5. Selection of suitable performance criteria, such as duty cycles and acceleration performance, on which to base results
6. Exercise of the computer program to generate comparative data for the study engines based on the selected performance criteria.

These elements are described in the following section. Throughout this section, reference is made to the OAP "Vehicle Design Goals - Six-Passenger Automobile" (Ref. 11). This document was intended to provide:

1. A common objective for prospective contractors
2. Criteria for evaluating proposals and selecting a contractor
3. Criteria for evaluating competitive power systems for entering first-generation system hardware.

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In it are guidelines defining maximum values for vehicle design parameters, such as propulsion system weight and volume, total vehicle weight, and emissions levels -- as well as selected vehicle performance criteria, including acceleration and grade speed requirements. Many of the vehicle data and performance requirements herein have been chosen from, or based on, the OAP goals, and are referenced accordingly.

Computer Program

General Description

The basic function of the automobile performance computer program is to estimate horsepower requirements, fuel consumption, and pollutant emissions for a given vehicle as it performs over a specified driving cycle. The program output is a time history of the vehicle's state which comprises parameters such as acceleration, speed, elapsed distance, and engine power output. At the end of each performance interval, average horsepower, fuel consumption in mpg and Btu/mi, and emissions of carbon monoxide (CO), unburned hydrocarbons (UHC), and nitrogen oxides (NO_x) in gm/mi are given. A computer program user's guide is given in Appendix I on page 169.

Performance Mode Options

The program offers nine performance mode options; the vehicle dynamics in all nine modes are constrained by road-to-tire adhesion, weight distribution, and available engine horsepower. Five of these modes allow the user to specify various constant-acceleration schedules and road grade profiles as functions of time or distance. With these options, for example, it is possible to simulate the California Driving Cycle, which consists of acceleration, deceleration, and constant-speed regimes, each at a constant magnitude of acceleration (mph/sec) and each for specified percentages of the total driving time. The sixth mode performs constant-horsepower acceleration, and may be used to determine maximum acceleration capability or top speed.

The remaining three modes command vehicle performance over programmed driving cycles, each corresponding to about 20 min of actual driving time. One of these cycles is the urban-type driving cycle which appears in the July 15, 1970 Federal Register*. The speed profile for this cycle (there are no grades) is shown in Fig. 23 with the associated horsepower requirements for a base-line 3850-lb vehicle, corresponding to the SSS-12 engine installation. Average power over the 1370-sec cycle is about 10.6 hp, including 1.3 accessory hp.** Another programmed

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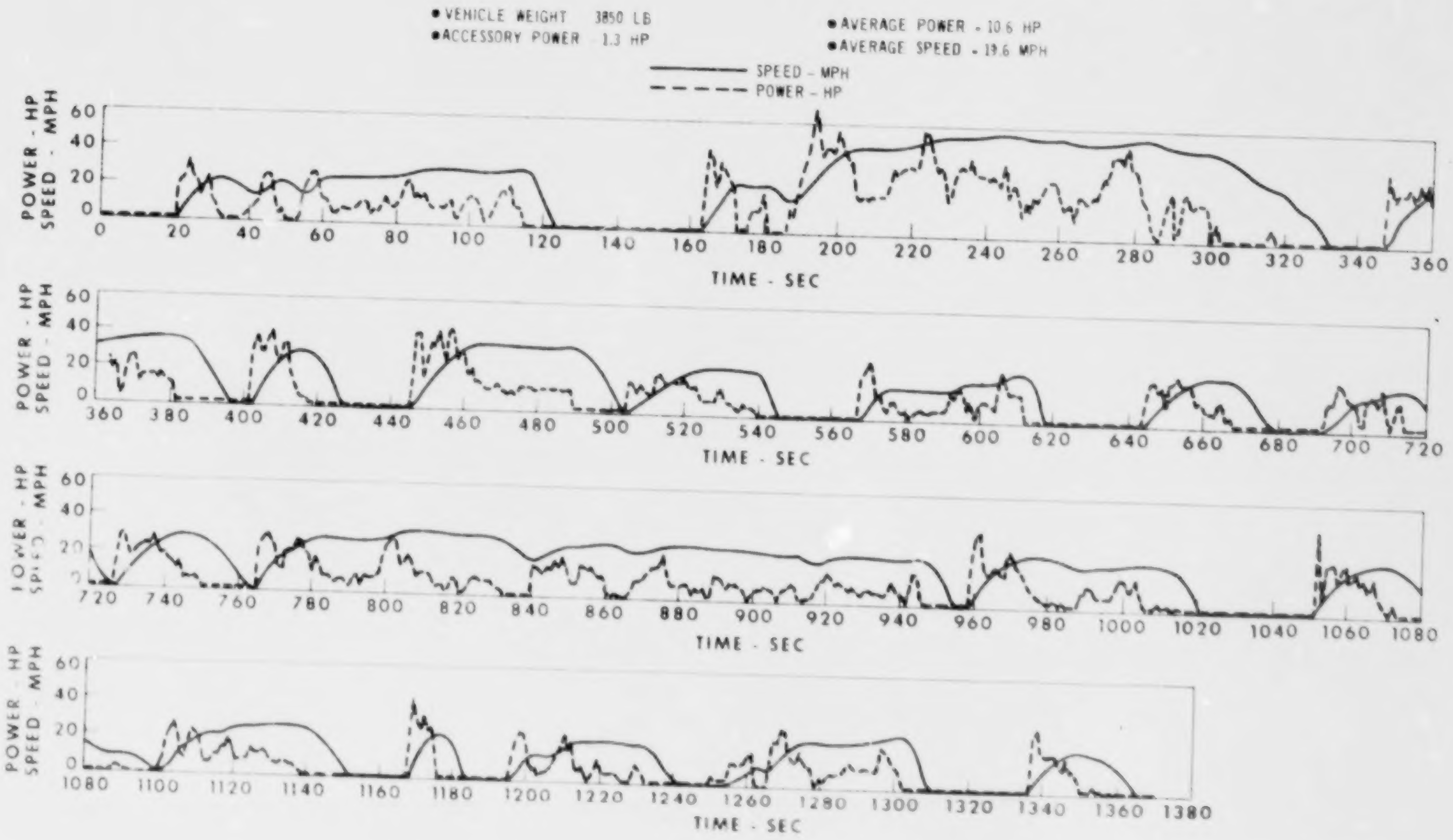
* This cycle will be referred to as the Federal Driving Cycle (FDC).

** This 1.3 hp accessory load is 10% of the 50-mph road load power required and is added to account for air conditioning as per Federal Driving Cycle requirements.

SPEED AND POWER PROFILES FOR JULY 15, 1970 FEDERAL REGISTER DRIVING CYCLE



driving cycle is the "UARL heavy-duty driving cycle" which was developed for engine design purposes, and which was intended to simulate heavy-duty, suburban driving. The speed and grade profiles of the UARL heavy-duty driving cycle are shown in Fig. 24, along with the horsepower required to operate the 3850-lb car at the specified speed and grade levels. Average horsepower for the 3850-lb vehicle over this 1080-sec cycle is 34.3 hp, including 1.3 accessory hp. Finally, a UARL "typical driving cycle" (Fig. 25) was developed which incorporated lower top speeds and lower acceleration levels than the UARL heavy-duty driving cycle, in order to show more typical fuel consumption for the average driver. This 1154-sec cycle represents country driving (differentiating it from the urban cycle represented by the FDC), and has an average power requirement of 29.3 hp, including 1.3 accessory hp for the 3850-lb SSS-12 car.

Input Data

The input and output data required by the program are summarized in Table III and are detailed below.

Vehicle

The vehicle input parameters serve to define the automobile geometrically and dynamically. Total vehicle weight is used, for simplicity, rather than individual component weights. An increase in engine weight of 25 lb, therefore, would be treated as a 25-lb increase in overall vehicle weight in the input data. The percent of the vehicle's weight on the driving wheels is included as an input, so that maximum available tractive force can be computed to determine wheel slippage conditions.

Frontal area and drag coefficient are input separately, rather than as a product. Rolling resistance coefficients are input for a quadratic resistance formula; specifically, the resistance is proportional to the product of the vehicle weight times a sum consisting of terms proportional to the zeroth, first, and second powers of the vehicle speed. Three resistance coefficients are therefore required, although any (or all) of them may be zero.

The total angular inertia of the wheels is treated as an input variable. This quantity is needed to calculate the additional power required to accelerate the wheels in rotation as the car accelerates in translation (linear motion), or in other words, to account for the effect of the apparent increase in the car's inertial mass. For all investigations considered here, this value was assumed to be included in the resistance formulas. The accessory horsepower loading is also included as part of the vehicle input. This parameter is the lumped sum of the horsepowers needed for the generator, fan, air conditioner, power steering, etc.

Fig. 24

SPEED, GRADE, AND POWER PROFILES FOR UARL HEAVY DUTY DRIVING CYCLE

- VEHICLE WEIGHT 3850 LB
- ACCESSORY POWER 1.3 HP
- AVERAGE POWER 34.3 HP
- AVERAGE SPEED 48.9 MPH

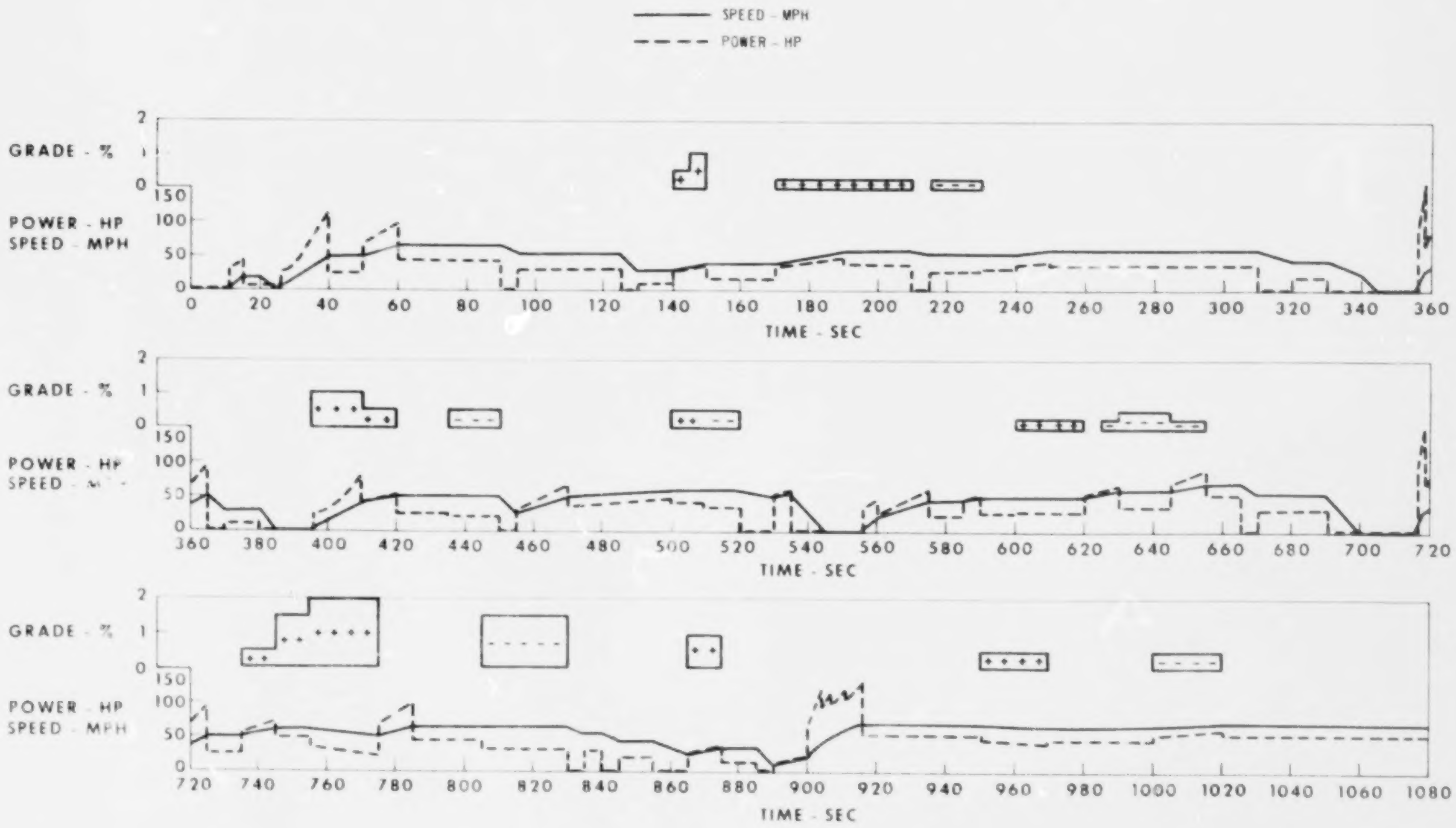


FIG. 25

SPEED AND POWER PROFILES FOR UARL TYPICAL DUTY DRIVING CYCLE

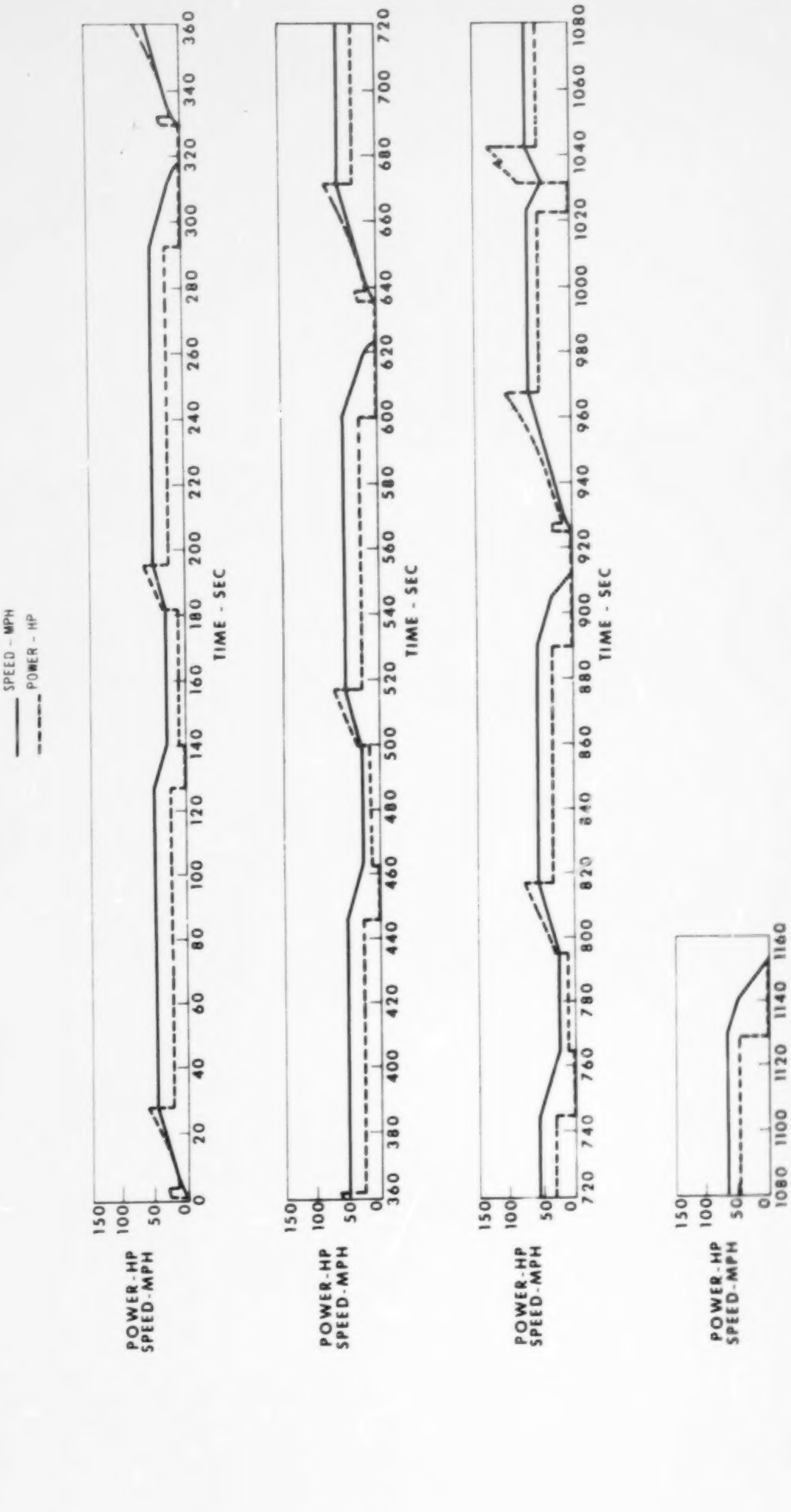


TABLE III

SUMMARY OF INPUT AND OUTPUT DATA

Input

Vehicle

1. Weight
2. Weight Distribution
3. Frontal Area
4. Drag Coefficient
5. Rolling Resistance Coefficients
6. Angular Inertia
7. Accessory Horsepower

Engine

1. sfc/Power Operating Line
2. Emissions/Power Curves
3. Maximum Horsepower

Drive Train

1. Transmission Efficiency Curve
2. Differential Efficiency
3. Axle Ratio

Road

1. Adhesion Coefficient
2. Grade

Output

Vehicle State

1. Acceleration
2. Speed
3. Distance
4. Fuel Flow
5. Emissions
6. Horsepower Breakdown
7. Total Horsepower

Total and Average Quantities

1. Total Emissions (grams)
2. Average Emissions (gm/mi)
3. Total Fuel Consumption (lb)
4. Average Fuel Consumption (mpg, Btu/mi)
5. Average Horsepower

Engine, Drive Train, and Emissions Data

Specific fuel consumption (sfc), transmission efficiency, and emissions data are input to the program in tabular form. One set of such data is required for each engine and transmission considered, although as many sets (engines and transmissions) may be input as desired. Numerical interpolation is performed by a self-contained subroutine. The fuel consumption data are input as a curve of sfc versus normalized horsepower*. This curve corresponds to the locus of points which define an "operating line" on the three-dimensional surface, or map, of sfc as a function of power (or torque) and rpm. In an actual automobile, the time history of the specific fuel consumption corresponds to a complicated "trajectory" on the sfc-power-rpm surface, the trajectory shape being determined by the driving cycle (performance requirements), transmission characteristics, and the driver. The operating line, in a sense, represents the mean path of many such trajectories projected on the sfc-power plane.

In addition to the sfc curve for each engine, it is also necessary to input the zero-output (idle) power slope and intercept for the corresponding fuel flow curve. This procedure is necessary to provide fuel flow data in the very low normalized-horsepower regimes, say below 0.0333 (corresponding to 5.0 hp for a 150-hp engine), where the slope of the sfc curve is go great as to make tabular input of data impractical. Although the fuel flow data are not directly amenable to use with normalized horsepower for data reduction, a technique was developed to extend the low-power output fuel flow data for a given engine to other engine sizes. This technique is documented in Appendix II on page 178.

A maximum (or rated) horsepower figure must also be input to constrain engine output and to define the normalized horsepower.

The transmission, like the engine, is represented by an operating line, in this case a single-valued function of efficiency versus percent output speed. In an actual transmission (the conventional hydrokinetic automatic, for example), overall efficiency would be computed as the product of that of the slipping device (the torque converter) and that of the mechanical gear reduction unit (the planetary gear sets). The gear efficiency is a function of the gear speed and load, but may be considered essentially constant (about 85%). The efficiency of a slipping device, such as a torque converter, is a function of its slip ratio, which in turn is a function of the vehicle speed, gear ratio, and acceleration load. The slipping efficiency, and hence the slip ratio, must be determined by a process of iteration, and involves the use of a set of converter performance curves unique to a particular torque converter. In addition, the transmission gear ratio must be known, necessitating knowledge of the transmission's shifting logic algorithm.

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* Normalized horsepower is defined as engine horsepower divided by maximum engine horsepower. The use of this parameter, rather than absolute horsepower, allows the engine to be scaled up or down (within limits) without additional data. For example, sfc data from a 150-hp engine might be used to generate fuel consumption curves for similar engines with horsepowers ranging from 125 to 175 by use of this "rubber engine" method. A further discussion of normalized horsepower is given in Appendix II.

Because of these effects, the rigorous determination of overall transmission efficiency, while by no means impossible, would involve a complicated mathematical and computational procedure whose cost would not be justified for an initial feasibility study of this type. Therefore, a single transmission operating line is used which corresponds to an average efficiency at each output speed, the average being taken over all gear and slip ratios; similarly, the rear axle is defined by its ratio and a single efficiency value.

The emissions data (in terms of an "emissions index," or lb of pollutant produced per 1000 lb of fuel consumed) is input as a function of normalized horsepower. Knowing normalized horsepower and fuel flow (lb/hr), the rate of pollutant emissions (lb/hr) can be calculated.

The use of an operating line, instead of a three-dimensional surface or map, to represent sfc and transmission data was dictated partly by a lack of complete and accurate data, and partly by a desire for simplicity, since the use of three-dimensional data for engine and transmission necessitates the introduction of a shifting logic algorithm for the transmission, as well as extensive efficiency data. Based on past experience with other, similar vehicle simulation programs, it is believed that the operating-line technique provides sufficient accuracy to make meaningful comparisons among various power train systems provided the operating line is selected with reasonable care, and provided a reasonably good transmission is selected. The method is, of course, much more accurate than using a single number to represent efficiency, particularly in the case of automatic transmissions, where efficiency drops off rapidly with increasing slip ratio (usually at low vehicle speeds). Future studies, such as engine design optimizations, will include the use of three-dimensional sfc and emissions maps, where appropriate, together with an appropriate transmission performance algorithm.

The road surface is represented by an adhesion coefficient and a percent grade. Other parameters, such as the heating value of fuel, ambient air density, and fuel density are listed internally and are easily changed.

Five other input parameters are required which control the reading format of the data, the computation interval width (sec), and the printing interval spacing (sec).

Output Data

The output data for each run are headed by a summary of the vehicle parameters for that run. The time history of the vehicle's state then follows. The history is composed of parameters such as acceleration, speed, elapsed distance, engine horsepower, and fuel flow. Other parameters include the rate of production of CO, UHC, and NO_x, in gm/hr, and a breakdown of the total horsepower into the amounts required to overcome aerodynamic drag and rolling and grade resistance, and the amount which is being used to accelerate the vehicle. Fuel consumption (lb), pollutants (gm), and engine output (energy in hp-sec) are continuously summed as the computations proceed*. At the end of each performance interval, these parameters are averaged, and the total and average values are printed out. Fuel consumption is given in lb, mpg, and Btu/mi, while pollutants are expressed in grams and gm/mi. A sample page of printout is shown in Fig. 26 for a simulated run over the Federal Register Driving Cycle (performance mode option 7). A print spacing of 100 sec between each line of output was used for this printout.

Vehicle Dynamics

This section details the equations and techniques which were used to develop the vehicle dynamics portion of the automobile performance computer program. The basic approach has been to compute the external forces acting on the vehicle due to gravity and aerodynamic and rolling resistance, and then, depending on the performance mode, to compute either the vehicle acceleration or the applied traction force. Dynamically, the vehicle is treated as a one-degree-of-freedom point mass (although weight distribution is taken into account in determining traction limitations). It is assumed that the driving force is transmitted equally through both driving wheels.

The following figure will serve to define the forces acting on the vehicle. Coordinates are defined parallel to the surface of the road, hence elapsed distance as computed by the program would correspond to the distance recorded on the vehicle's odometer. All units are lb, ft, and sec, unless otherwise noted.

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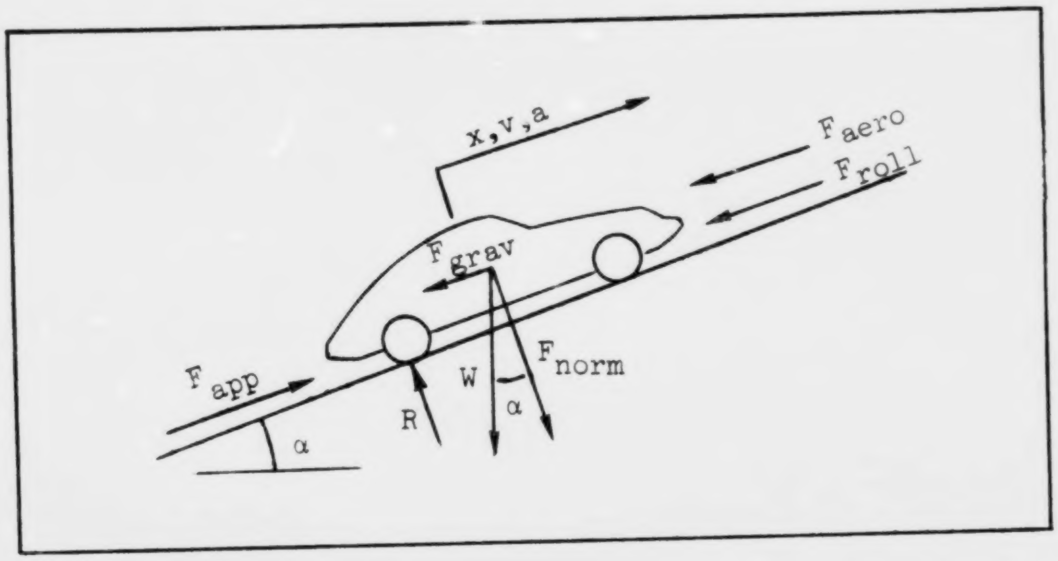
* For example, the fuel consumption for each computation interval (usually one second) is taken as the product of the instantaneous fuel flow (lb/sec) and the computation interval width (sec). This amount is then added to the previously stored value of the total fuel consumption (lb).



The definition of percent grade can be employed to determine the gravitational forces acting on the vehicle, as follows:

$$F_{grav} = W\ (\%\ \text{Grade}/100) \tag{3}$$

$$\sin\alpha = F_{grav}/W = \%\ \text{Grade}/100 \tag{4}$$

$$F_{norm} = W\cos\alpha \tag{5}$$

where F_{grav} is the gravity force acting on the vehicle in direction of motion, W is vehicle weight, % Grade represents percent grade, α is the grade angle relative to horizontal, and F_{norm} is the gravity force acting on vehicle normal to the direction of motion.

The normal reaction force on the vehicle's driving wheels (shown as the rear wheels in the sketch) may be determined as

$$R = F_{norm}\ (\%\ \text{Driving}/100), \tag{6}$$

where R is the normal reaction force on the driving wheels, and % Driving represents the percent of vehicle's weight on driving wheels. The maximum force, F_{max}, that may be applied to the driving wheels before slippage occurs is

$$F_{max} = \mu R, \tag{7}$$

where μ is the adhesion coefficient.

FIG. 26

SAMPLE COMPUTER PRINTOUT

```
*** VEHICLE PERFORMANCE ***

     VEHICLE PARAMETERS

* ENGINE 2 (SIMPLE CYCLE,SINGLE SHAFT/FREE TURBINE)
* TRANSMISSION B (8-SPEED AUTOMATIC)
WEIGHT(LB)=                3850.
EQ. WHEEL WEIGHT(LB)=         0.
DRAG COEFFICIENT=           .480
FRONTAL AREA(FT**2)=        25.0
WHEEL RADIUS(FT)=          1.100
AXLE RATIO=                3.000
P-CNT WT ON REAR WHEELS=    50.0
ACCESSORY HORSEPOWER=        1.3
RATED HORSEPOWER=          150.0


INTERVAL PARAMETERS ** MODE 7

PER CENT GRADE=               .0
FRICTION COEFFICIENT=       1.00
ACCEL REQD(MPH/SEC)=         .00


                                               ACC    AERO    ROLL   GRADE    TRAC   TOTAL     FUEL   * * * POLLUTANTS * * *
  TIME     ACC   SPEED     DIST     RHO     POWER   POWER   POWER   POWER   POWER   POWER     FLOW      CO      UHC      NOX
 (SEC) (MPH/S)   (MPH)     (MI)              (HP)    (HP)    (HP)    (HP)    (HP)    (HP)  (LB/HR)       (GRAMS/HOUR)
    .0     .00      .0     .000     .00        .0      .0      .0      .0      .0     1.3     13.5  51.392   11.813    6.338
 100.0     .30    29.9     .497     .07       4.2     2.2     5.0      .0    11.4    15.1     22.5  35.656   12.736   14.726
 200.0    1.70    42.0     .911     .22      33.4     6.0     7.3      .0    46.8    57.9     42.7  29.967    7.679   72.807
 300.0    -.30    49.2    2.359     .05      -6.9     9.7     8.8      .0    11.7    15.4     22.7  35.611   12.726   14.984
 400.0     .00     -.0    2.994     .00        .0     -.0      .0      .0     -.0     1.3     13.5  51.392   11.813    6.338
 500.0   -2.50    12.5    3.580    -.19     -14.6      .2     2.0      .0   -12.5     1.3     13.5  51.392   11.813    6.338
 600.0     .40    21.1    3.948     .08       3.9      .8     3.4      .0     8.1    11.3     19.6  35.406   12.776   11.873
 700.0    1.30    14.3    4.238     .15       8.7      .2     2.3      .0    11.2    16.1     23.0  35.444   12.680   15.459
 800.0    1.00    28.7    4.777     .14      13.4     1.9     4.3      .0    20.1    25.7     27.0  32.367   11.399   22.997
 900.0    -.10    26.9    5.561     .03      -1.3     1.6     4.5      .0     4.8     7.1     16.6  35.741   12.401    9.114
1000.0     .70    24.2    6.177     .11       7.9     1.2     4.0      .0    13.1    17.2     23.3  34.934   12.521   16.180
1100.0     .50      .3    6.507     .08        .1      .0      .0      .0      .1     5.5     15.7  37.213   12.321    8.293
1200.0    1.60    10.9    6.859     .18       8.2      .1     1.7      .0    10.0    16.5     23.1  35.272   12.628   15.762
1300.0     .20    28.8    7.292     .06       2.7     2.0     4.8      .0     9.4    12.7     20.9  35.884   12.943   13.046
1370.0     .00     -.0    7.474     .00        .0     -.0      .0      .0     -.0     1.3     13.5  51.392   11.813    6.338

            ***** FUEL AND EMISSIONS DATA *****

FUEL USED (LB) =    7.126      MILEAGE (MPG) =    7.342
CO (GM)        =   15.736      CO (GPM)      =    2.105
UHC (GM)       =    4.528      UHC (GPM)     =     .606
NOX (GM)       =    4.839      NOX (GPM)     =     .647
BTUS PER MILE  =   17543.

AVE HORSEPOWER =   10.596
```

The aerodynamic resistance force, F_{aero}, is defined by

$$F_{aero} = \tfrac{1}{2}\rho C_D A V^2, \tag{8}$$

where ρ is the ambient air density, C_D is the drag coefficient, A is the frontal area, and V is the speed. The rolling resistance force, F_{roll}, is defined by

$$F_{roll} = W_{norm}\,(a_o + a_1 V + a_2 V^2), \tag{9}$$

where a_o, a_1, a_2 are the rolling resistance coefficients.

The vehicle acceleration is then given by the relation

$$m'a = \Sigma F = F_{app} - F_{grav} - F_{aero} - F_{roll}, \tag{10}$$

where m' is the apparent inertial mass, a is the acceleration, and F_{app} is the traction force applied at the driving wheels. The apparent inertial mass m', which is equal to the gravitational mass of the vehicle plus the apparent additional mass due to the rotary inertia effect of the wheels, assuming no slippage, is defined by

$$m' = W/g + 4I/r^2 + I_e R^2/r^2 \; {}^{*} \tag{11}$$

where g is the acceleration due to gravity, I is the rotational inertia of one wheel (including tire), I_e is the effective rotational inertia of the engine, R is the total gear reduction ratio between the engine and the wheels, and r is the rolling radius of the wheel.

---

[*] The contribution of the engine to the equivalent mass of the vehicle can be significant (above 30%), but this occurs mainly in the lowest gear ratio(s) where R is large, the effect being a reduction in maximum acceleration capability. The last term in Eq. (1), however, is applicable only when the engine is directly geared (or otherwise connected) to the wheels without slippage of the engine relative to the wheels, since slippage reduces the engine inertia effect. The exact determination of the effects of engine mass on equivalent vehicle mass is beyond the scope of this analysis, since it depends on detailed specification of slip ratio, gear ratio, idle speed, etc. The contribution of the engine inertia term was neglected in this analysis, and it is believed that the error introduced is smaller than the uncertainties regarding the transmission efficiency, average power assumption, and traction coefficient.

The horsepower applied at the driving wheels is defined by

$$HP_{app} = F_{app}\, V/550. \tag{12}$$

The total engine horsepower, HP_{eng}, is defined as the applied horsepower at the wheels plus transmission and rear axle losses plus the accessory horsepower:

$$HP_{eng} = HP_{app}/\eta_a \eta_t + HP_{acc}, \tag{13}$$

where η_a is the rear axle efficiency, η_t is the transmission efficiency, and HP_{acc} is the accessory horsepower. The first term on the right is equal to the applied horsepower plus transmission and rear axle losses, and is also called flywheel horsepower.

There are two separate classes of motion to be treated. In the first, vehicle acceleration is specified and the engine horsepower needed to produce that acceleration is to be determined. In the second class, the engine horsepower is specified and the vehicle acceleration must be found. However, for each class of motion, the vehicle performance may be constrained by traction limitations such that the specified performance may not be obtainable.

Class 1 - Acceleration Specified ($a = a_{req}$), where a_{req} is the acceleration required by the driving cycle.

Since the maximum driving force which can be transmitted to the road is F_{max} (Eq. (7)), the applied force must be the lower of F_{max} or the force required to produce an acceleration of a_{req}.

From Eq. (10)

$$F_{req} = m' a_{req} + F_{grav} + F_{aero} + F_{roll}, \tag{14}$$

where F_{req} = required tractive force, and

$$F_{app} = \min \{F_{req}, F_{max}\}. \tag{15}$$

If $F_{app} = F_{max}$ (traction-limited acceleration), the required acceleration cannot be produced, and the actual acceleration is solved for from

$$m'a = F_{max} - F_{grav} - F_{aero} - F_{roll}. \tag{16}$$

In either case, applied and engine horsepowers are then obtained from Eqs. (12) and (13), respectively.

Class 2 - Horsepower Specified ($HP_{eng} = HP_{com}$), where HP_{com} is the commanded engine horsepower.

For the performance mode in which engine output power (HP_{com}) is commanded, it is necessary to solve for the power at the wheels which this engine horsepower produces. If this power is greater than the maximum power which can be transmitted to the road due to the F_{max} limit, an engine power is solved for which just produces a force of F_{max}.

The power available at the wheels corresponding to an engine output of HP_{com} is obtained from Eq. (13).

$$HP_{avail} = \eta_a \eta_t (HPcom - HP_{acc}), \tag{17}$$

where HP_{avail} = horsepower available at driving wheels. The maximum power which can be applied to the wheels is found by substituting Eq. (7) into Eq. (12)

$$HP_{max} = F_{max} V/550, \tag{18}$$

where HP_{max} is the maximum tractive horsepower.

The applied power at the wheels is taken as the lower of HP_{avail} or HP_{max}, which is expressed as

$$HP_{app} = \min \{HP_{avail}, HP_{max}\} \tag{19}$$

if $HP_{app} = HP_{max}$, engine horsepower is obtained by substituting HP_{max} into Eq. (13). The resulting HP_{eng} will, of course, be less than or equal to HP_{com}. The applied force is solved for from Eq. (12) and the corresponding acceleration from Eq. (10).

The actual computation of the vehicle speed and position is performed by simple numerical integration, i.e.,

$$\Delta V = a\Delta t, \tag{20}$$

where a is the acceleration (assumed constant), Δt is the time increment, and ΔV is the speed increment. Similarly,

$$\Delta X = V\Delta t + a\Delta t^2/2, \tag{21}$$

where ΔX is the distance increment.

At the beginning of each computation interval, the speed is known and the acceleration is either known or calculated. Required horsepower, fuel consumption, and emissions rates are then computed and, along with the acceleration, are assumed to remain constant during the interval Δt. Speed and distance at the end of the interval are computed by adding the incremental values obtained from Eqs. (20) and (21) to the known values of speed and distance at the beginning of the interval, the process being repeated until the end of the driving cycle. (The vehicle acceleration history is therefore a type of step function, and the difference between this step function and the smooth curve which it approximates can be brought to any desired degree of smallness by suitably defining the step size.) For average driving cycles, interval sizes of about one second are small enough to produce acceptable accuracy in fuel consumption and emissions calculations because of the low rates of change of acceleration.

Performance Analysis

Definition of Base-Line Vehicles

Vehicles were sized according to the guidelines given in Ref. 11. The reference vehicle is an Otto-cycle engined automobile representative of a 1970 standard-sized 6-passenger sedan, which weighs 4000 lb, including weight of fuel. Curb weight was taken as the sum of a 2700-lb bare weight (vehicle less propulsion system) plus 1300 lb for engine, transmission, and associated propulsion system weights. Vehicle test weight (W_{test}) was defined as curb weight plus 300 lb, and vehicle gross weight (W_{gross}) as curb weight plus 1000 lb, where the 1000-lb load simulates a full load of passengers and baggage. Weight distribution was assumed to be 50%-50% (front/rear).

The estimated vehicle weights are presented in Table II (page 40). The component weight apportionments of the reference vehicle are calculated from percentages reported by Hoffman (Ref. 12). Engine weights of the gas turbine engines are those estimated from the design portion of the study. The engine weight of the OC-76 engine was assessed as 10% more than the OC-70, to compensate for an assumed 10% power loss as a result of the combination of lowering compression ratio to burn unleaded gasoline to avoid catalyst poisoning, and exhaust gas recirculation to lower the equivalence ratio and NO_x formation. An additional 50 lb is added to the OC-76 exhaust system for pollution control equipment, which might include catalytic mufflers and/or reactors. Exhaust system weights are increased 50% over the OC-70 for the gas turbines, based on an increase of 100% (35 lb) for the assumed piping increases, and a decrease of 67% (10 lb) for the assumed muffler requirements.

The input transmission weights for the gas turbine engines were based on a manufacturer's estimate for the GE HMT transmission for the single-shaft engine, and the contractor's judgment that 25 lb of weight would be saved by eliminating the torque converter in the free-turbine versions. Fuel and tank weights were based on comparable fuel consumption over the FDC, where the OC-70 vehicle is assumed to have the 200-mile range for that cycle as specified in Ref. 11.

The OAP guidelines (Ref. 11) also provided standard specifications for the air drag and rolling resistance calculations. The aerodynamic drag is given by the relation

$$\text{Drag} = \tfrac{1}{2}\rho C_D A V^2, \qquad (22)$$

where A = vehicle frontal area (ft^2)

C_D = drag coefficient based on frontal area

V = speed (fps)

ρ = ambient air density ($slugs/ft^3$).

OAP specifies that the product of the drag coefficient and the frontal area, $C_D A$, shall have a value of 12 ft^2. (This is equivalent to a frontal area of 25 ft^2 and a drag coefficient of 0.48.) The OAP rolling resistance equation is given as

$$\text{Resistance (lb)} = W[.015 + (2.15 \times 10^{-5}V) + (1.85 \times 10^{-7}V^2)] \qquad (23)$$

where W = vehicle weight (lb).

A complete summary of the parameters used to define the base-line vehicle is given in Table IV, along with the selected values of pertinent physical constants. Wheel rolling radius was assumed to be 1.1 ft, and accessory power (generator, power steering, air conditioner, etc.) was taken as 1.3 hp for the three driving cycles (FDC, UARL heavy-duty and UARL typical), and 4.0 hp for all other cases. The rear axle ratio was computed to be 3.0 (for a transmission output speed of 3500 rpm at 100 mph), and the axle efficiency was assumed to be 0.95. A road-tire adhesion coefficient (the maximum coefficient of friction between the vehicle tire and road surface) of 1.0 was selected, and the angular inertia of rotating parts was assumed to be included in the overall values in the rolling resistance equation above.

Fuel Consumption

As noted earlier, the fuel consumption data for each engine are in the form of an operating line, which is a single curve of fuel consumption versus normalized horsepower. The fuel consumption data for the 150-hp base-line engines are presented in the curves of Figs. 27 and 28, which show fuel flow and sfc as functions of engine horsepower for the five engines. All fuel consumption curves are estimated for standard-day (59 F, sea level) conditions. Higher ambient temperatures and altitudes would degrade gas turbine performance more severely than Otto-cycle performance. Note that the fuel flow and sfc curves for the single-shaft and free-turbine versions of the simple-cycle engine are identical.

The sfc operating line curves of Fig. 28 are shown with a double-scale abscissa, one for engine power, which corresponds to the curves of Fig. 27, and one for normalized power. Data points from Fig. 28 (sfc vs normalized hp) were used, in tabular form, as computer input.

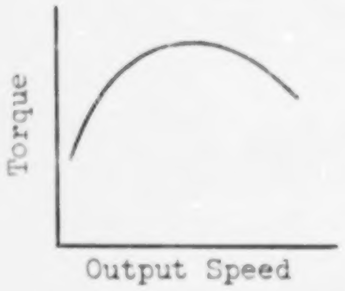
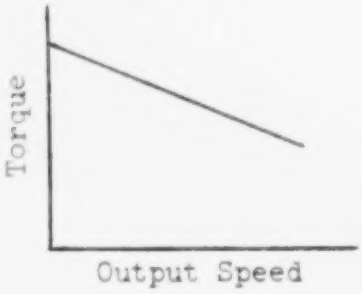
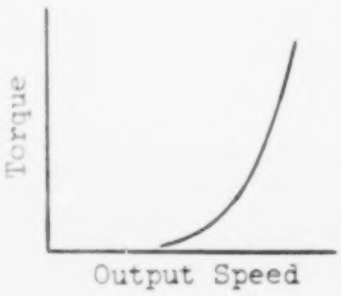
Fuel flow and sfc data for the SSS-12, SFT-12, RSS-7 and RFT-4 engines were taken from Figs. 9, 12, 15, and 18, respectively. The fuel flow and sfc data for the OC-70 engine were derived from Ref. 13 and include fan power losses. The data of Figs. 27 and 28 are near-optimum for each power level; that is, for a given power level, the minimum sfc or fuel flow was chosen from the sfc-power-rpm map.

Transmission Data

Four types of automatic transmissions were selected and matched to the engines whose power-speed characteristics they suited best. These transmissions are defined by their weight and efficiency-output speed curves, as described in the following discussion.

Transmission matching was done on the basis of engine torque-speed characteristics, typical examples of which are shown in the sketches below (see also Figs. 9, 12, 15, and 18). The curve in each figure represents the maximum engine torque available as a function of output speed, and the engine may operate anywhere under this curve.

Torque-Speed Characteristics

Otto-Cycle	Free-Turbine	Single-Shaft
		

The Otto-cycle engine, which has a fairly flat torque curve over most of its operating speed range, was matched to a conventional 3-speed automatic transmission with torque converter. The steady-state efficiency of this transmission was assumed to be 85%.

The free-turbine engines (both simple-cycle and regenerated versions) are characterized by a torque curve which rises linearly as speed decreases, and were matched to a 4-speed automatic transmission which was assumed to weigh 135 lb. This transmission utilizes a 4-speed gear set and can be visualized as using the power turbine of the engine in place of a clutch or hydraulic coupling. The converter-like slipping losses in the power turbine are then assumed to be reflected in decreasing efficiency of this transmission at low output speeds in order to account for the greater fuel consumption expected from the free turbine at those conditions. This assumption, while physically inaccurate, is justifiable for computer programming purposes, since the total engine power output is constrained by traction limits at the low speed end where the assumption is used. The steady-state efficiency of this device was assumed to be four percentage points above the conventional transmission, i.e., 89%, which is justified on the basis of the elimination of the fluid coupling action of the torque converter at cruising speeds.

The single-shaft engines present the most severe transmission requirements of the engines considered because of their relatively narrow band of useful engine torque (the area under the torque curve). This characteristic makes these engines

TABLE IV

SUMMARY OF BASE-LINE VEHICLE PARAMETERS

Parameters	Value	
Weight	See Table IV	
Weight distribution	50%-50%	
Frontal area	25 ft^2	
Drag coefficient	0.48	
Rolling resistance coefficients	a_o = 0.015 lb/lb	See Fg. 63
	$a_1 = 2.15 \times 10^{-5}$ lb/lb-fps	
	$a_2 = 1.85 \times 10^{-7}$ lb/lb-fps^2	
Wheel radius	1.1 ft	
Accessory power		
All driving cycles	1.3 hp*	
Acceleration performance and constant-speed fuel economy	4.0 hp	
Rear axle ratio	3.0	
Rear axle efficiency	0.95	
Adhesion coefficient	1.0	

SELECTED PHYSICAL CONSTANTS

Acceleration of gravity	32.2 ft/sec^2
Ambient air density**	
T = 60 F	0.0023723 slug/ft^3
T = 85 F	0.0022634 slug/ft^3
Fuel density	
Gasoline	6.2 lb/gal (0.161 gal/lb)
Turbine fuel	7.0 lb/gal (0.143 gal/lb)
Lower heating value (LHV) of fuel	
Gasoline	18,650 Btu/lb
Turbine fuel	18,400 Btu/lb

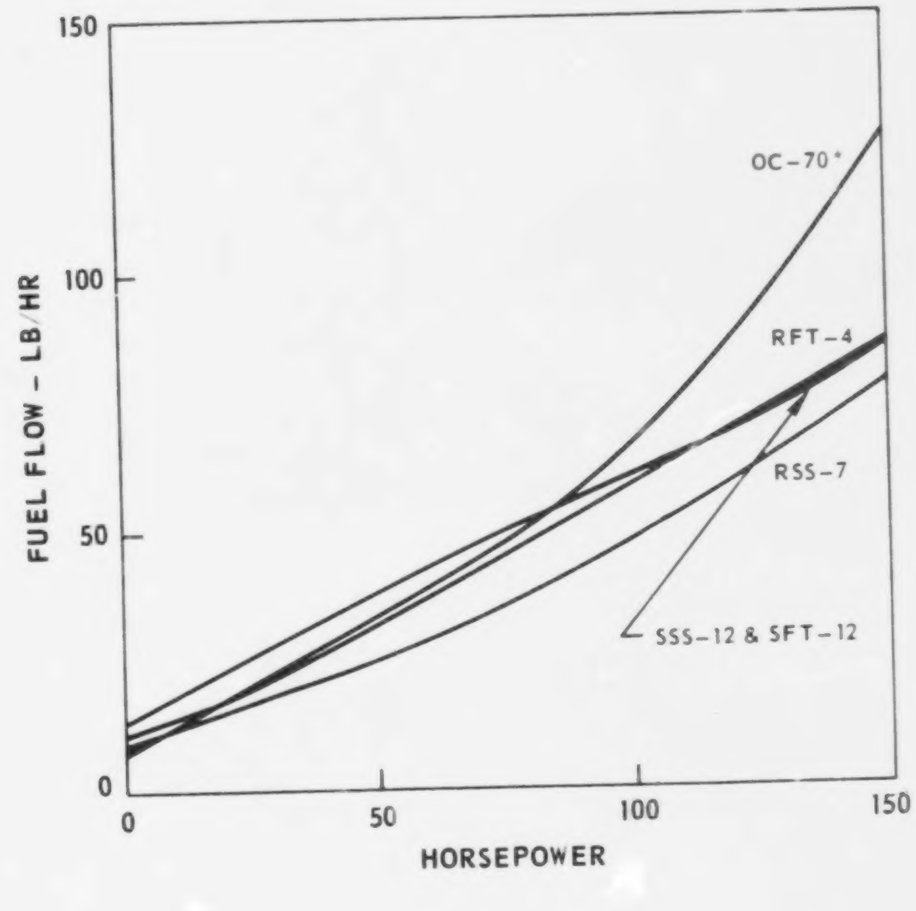
---

* 1.3 hp is the dynomometer power added at 50 mph to provide for air conditioned cars tested on the FDC.

** 1962 US Standard Atmosphere corrected to ambient temperature.

FUEL FLOW VERSUS HORSEPOWER

STANDARD DAY

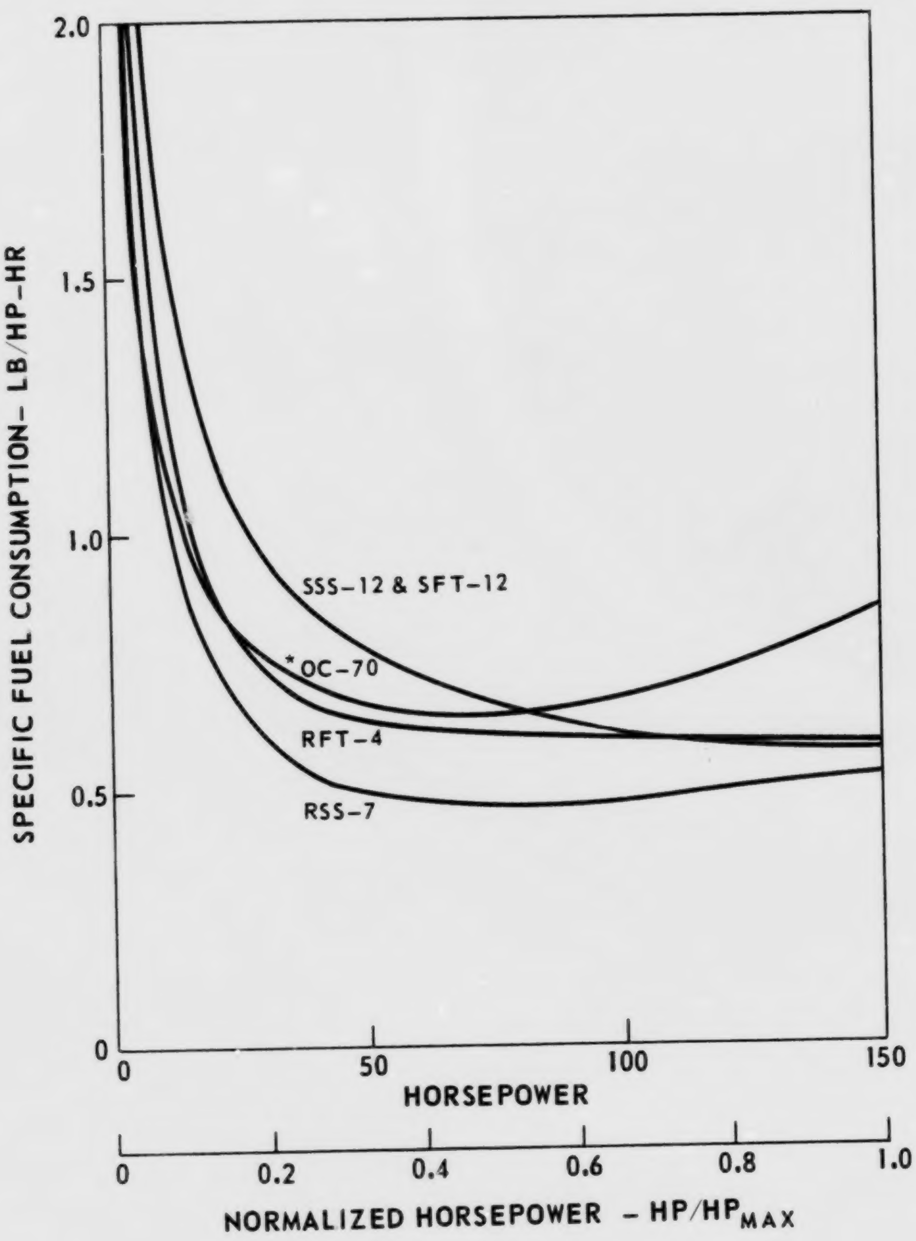


* NET, INCLUDING FAN LOSS (DERIVED FROM REF. 12)

FIG. 28

SPECIFIC FUEL CONSUMPTION VERSUS HORSEPOWER

STANDARD DAY



*NET, INCLUDING FAN LOSS (DERIVED FROM REF. 12) best suited to mechanical or hydromechanical transmissions with many speeds or with continuously variable speed ratios (infinitely variable), such as any of many electric drives (Refs. 12, 14, 15, and 16). For this study, the single-shaft engines were matched to an 8-speed automatic transmission suggested by Borg-Warner, who acted as consultants in the drive-line area. This transmission drives through a controlled slipping clutch to a 4-speed gear set through a 2-speed planetary splitter, and is assumed to weigh 185 lb. The steady-state efficiency of this drive system was assumed to be between the 3-speed and the 4-speed automatics discussed above, at a value of 87%. This assumption is justified as a result of a 4% increase due to eliminating the fluid coupling inefficiency of the torque converter (as in the 4-speed transmission), and a 2% decrement in efficiency due to the additional gear mesh ascribed to the splitter, for a resultant net improvement of 2% in efficiency when compared with the conventional 3-speed automatic transmission.

The General Electric continuously variable hydromechanical transmission (HMT) was selected by UARL as an alternative transmission for evaluation with the single-shaft engines. This transmission, although having a lower overall efficiency than the 8-speed device, allows full engine power to be utilized, because of its automatically controlled, infinitely variable characteristics. Efficiency data for this transmission were taken from Ref. 10 and data supplied by the manufacturer.

A summary of engine and transmission types is given in Table V. Curves of estimated transmission efficiency versus percent output speed are shown in Fig. 29. As explained previously, they are single-valued functions of the percent output speed, representing average values taken over all gear and input-output speed ratios. Percent output speed, as used to define the curves of Fig. 29, indicates the ratio (expressed as a percent) of transmission output speed to maximum output speed, which was taken as 3800 rpm. All transmissions were assumed to have zero efficiency at zero output speed, because the power transmitted is zero (infinite slip ratio), and were assumed to reach a constant value of efficiency at about 30% output speed or about 30 mph (with one exception which is discussed later). This means that the car is assumed to operate at about road-load performance levels above 30 mph, and that full engine power (less the constant loss percentage) is available at all speeds above 30 mph. The degradation of vehicle performance under heavy loads above 30 mph (say on a full-power, 0-60 mph run) due to this constant-efficiency assumption is accounted for by reducing maximum available engine power on heavy-load performance runs. This technique is discussed in a later section.

Emissions

General Discussion

The overall lean fuel-air ratios, and the absence of quenching on cool walls in the gas turbine engine result in more complete combustion and, consequently, lower levels of UHC and CO emissions, compared with a typical Otto-cycle engine.

TABLE V

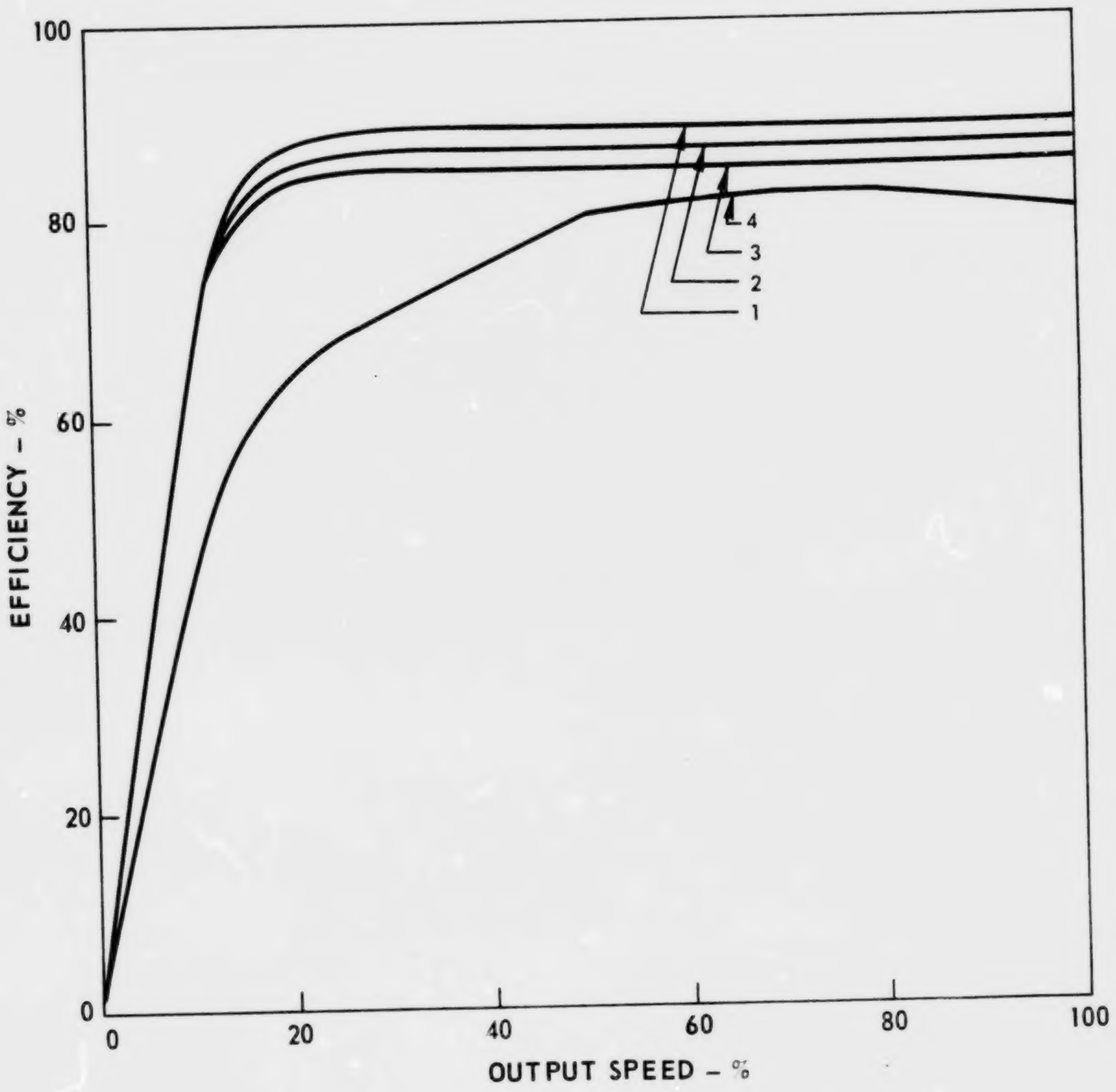
ENGINE AND TRANSMISSION TYPES

ENGINE			TRANSMISSION	
Designation	Type	Computer Code	Type	Computer Code
RFT-4	Basic Regenerated, Free-Turbine	1	4-speed	1
SSS-12	Simple-Cycle, Single-Shaft	2	8-speed	2
			HMT	4
SFT-12	Simple-Cycle, Free-Turbine	2	4-speed	1
RSS-7	Advanced Regenerated, Single-Shaft	3	8-speed	2
			HMT	4
OC-70	Otto-Cycle	4	3-speed	3

FIG. 29

TRANSMISSION EFFICIENCY VERSUS PERCENT OUTPUT SPEED

TRANSMISSION	DESCRIPTION
1	4-SPEED
2	8-SPEED
3	3-SPEED
4	HYDROMECHANICAL



In the gas turbine, the UHC's are of two major types: the first is formed by those hydrocarbons which pass through the combustion chamber unchanged; the second consists of those which are emitted as a result of incomplete combustion in the low-temperature regions of the combustor. The formation of both types, as well as the formation of CO, takes place in the outer zones of the combustion chamber, where the air injected to cool the liner walls has lowered the local temperature. The pollutant emissions are characteristically highest at idle conditions for UHC and CO and, as the engine power output increases and combustion improves, the emissions of UHC and CO for all practical purposes become negligible. (It must be noted that the characteristics which tend to increase combustion efficiency and promote proper combustion, i.e., oxidation, generally tend to decrease the emission of UHC and CO.)

The formation of NO_x is caused by the oxidation of nitrogen by free oxygen radicals. Measurements to date have indicated that the rate of this formation is a strong function of both temperature and residence time within the high-temperature zones of the combustor, but is a weak function of pressure ratio, at least for practical gas turbine cycle pressures (Ref. 17). Measurements show NO_x levels for gas turbine engines to be distinctly lower than those expected for equilibrium conditions (Ref. 18). The NO_x is essentially frozen at the combustor exit, and concentrations remain the same during the expansion process. Among the reasons postulated for this low NO_x concentration in the gas turbine engine are the rapid quenching of combustion products with the excess air introduced into the combustor for cooling, and the rapid average transit times of the gases through the primary zone. Since most of the NO_x is apparently formed within the combustor primary zone, the formation of NO_x is probably determined by the residence time which is a function of recirculation within this zone and by the peak temperatures of this zone, which are, in turn, functions of both combustor inlet temperature and equivalence ratio.

In the estimates of emissions data, particular attention was given to the oxides of nitrogen, as opposed to UHC and CO, because (1) oxides of nitrogen are considerably more toxic than either UHC or CO, and thus more undesirable, and (2) CO and UHC emissions can be reduced relatively more easily (by promoting complete combustion) than control of NO_x formation (which is a direct result of the combustion process). Furthermore, compliance with CO and UHC standards has already been demonstrated by a gas turbine automobile (Ref. 19).

The estimation of emissions data is hampered by the general lack of reliable information, and inasmuch as the evaluation of emissions characteristics was not a primary study objective, no independent tests or development programs were performed to verify the levels estimated. Available estimates of gas turbine emissions range from those associated with very large aircraft engines down to those of small regenerative automotive gas turbines. Moreover, because of the very small concentration of oxides of nitrogen in typical gas turbine exhaust streams, measurements are difficult to make and are subject to considerable error. Consequently, considerable scatter exists in the reported data. For example, Sawyer (Ref. 18) has reported little correlation between NO_x formation and overall equivalence ratio for various gas turbines*. Finally, analytical models based on kinetic theory are not entirely satisfactory because of the difficulty of predicting residence time and local fuel-air ratios. These residence times are vital in predicting the amount of oxides of nitrogen formed in a gas turbine combustor, since gas turbine nitrogen oxide formation is typically only a small percentage of those values predicted by equilibrium theory, while local fuel-air ratios are strongly influenced by liquid droplet evaporation.

Emissions Data

The emissions data assumed for purposes of deriving emissions estimates for automobiles powered by the engines under consideration are shown in Figs. 30, 31, and 32 for NO_x (oxides of nitrogen), CO (carbon monoxide), and UHC (unburned hydrocarbons), respectively. These data are presented in terms of an emissions index (EI, defined as pounds of pollutant produced per 1000 lb fuel consumed) versus normalized horsepower. These data, like sfc, have been assumed to be single-valued functions of normalized power, representing a mean EI at each power level. In Fig. 30, NO_x emissions data for the simple-cycle engines were taken from Ref. 20, as measured on the UACL PT6A-27 gas turbine, and the NO_x EI for the regenerated engines was derived from Ref. 21. The NO_x, CO, and UHC EI data for the OC-70 engine were taken from the curves of Ref. 22.

The CO and UHC EI data for the gas turbines are UARL estimates of performance which could be obtained from either improved fuel pressurizing nozzles or state-of-the-art air blast atomizing nozzles. The EI estimates for CO and UHC shown for all gas turbine engines at full power are typical of measured aircraft gas turbine engine emissions while the corresponding estimate shown for the regenerated engine at idle is typical of designs using air blast atomizing nozzles, such as the GM GT 309 reported in Ref. 21. The EI estimates for CO and UHC at idle for the simple-cycle engines, however, are not typical of current aircraft engine tests, but rather represent an increase, by a factor of 1½ to 2, in those estimated for the regenerated engines at idle. This estimate accounts for the fact that present fuel pressurizing nozzles used on aircraft gas turbines do not produce the finely atomized fuel spray patterns at idle essential for a low-pollution gas turbine engine, but recognizes the inherent advantage of the regenerated engine's higher combustor inlet temperature in promoting more complete combustion.

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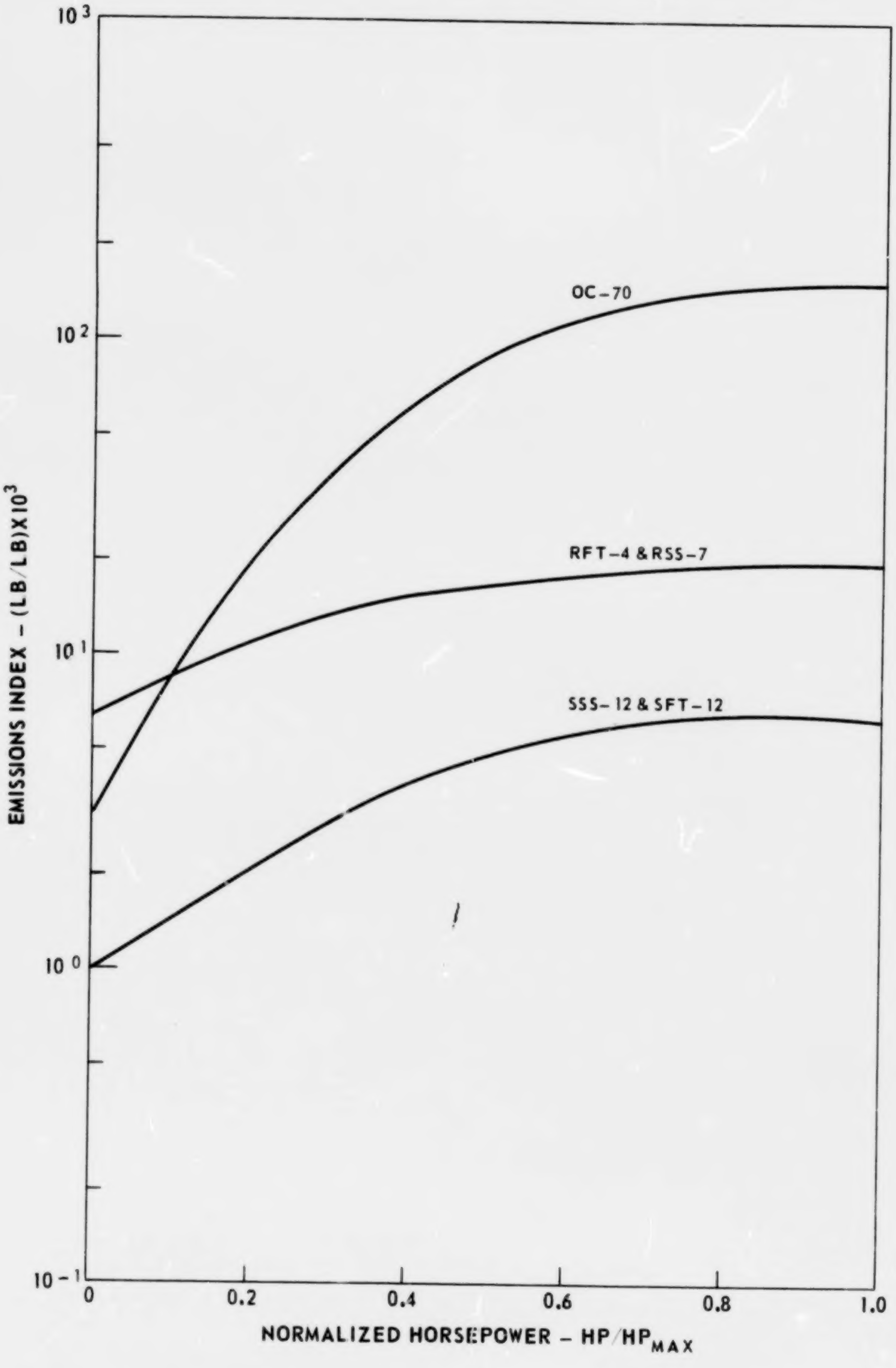
* Although NO_x formation is a strong function of primary zone (as opposed to overall) equivalence ratio, values for localized equivalence ratios in specific engines are generally not known.

EMISSIONS INDEX VERSUS NORMALIZED HORSEPOWER NITROGEN OXIDES (NO_X) (EXPRESSED AS NO_2)



EMISSIONS INDEX VERSUS NORMALIZED HORSEPOWER
CARBON MONOXIDE (CO)

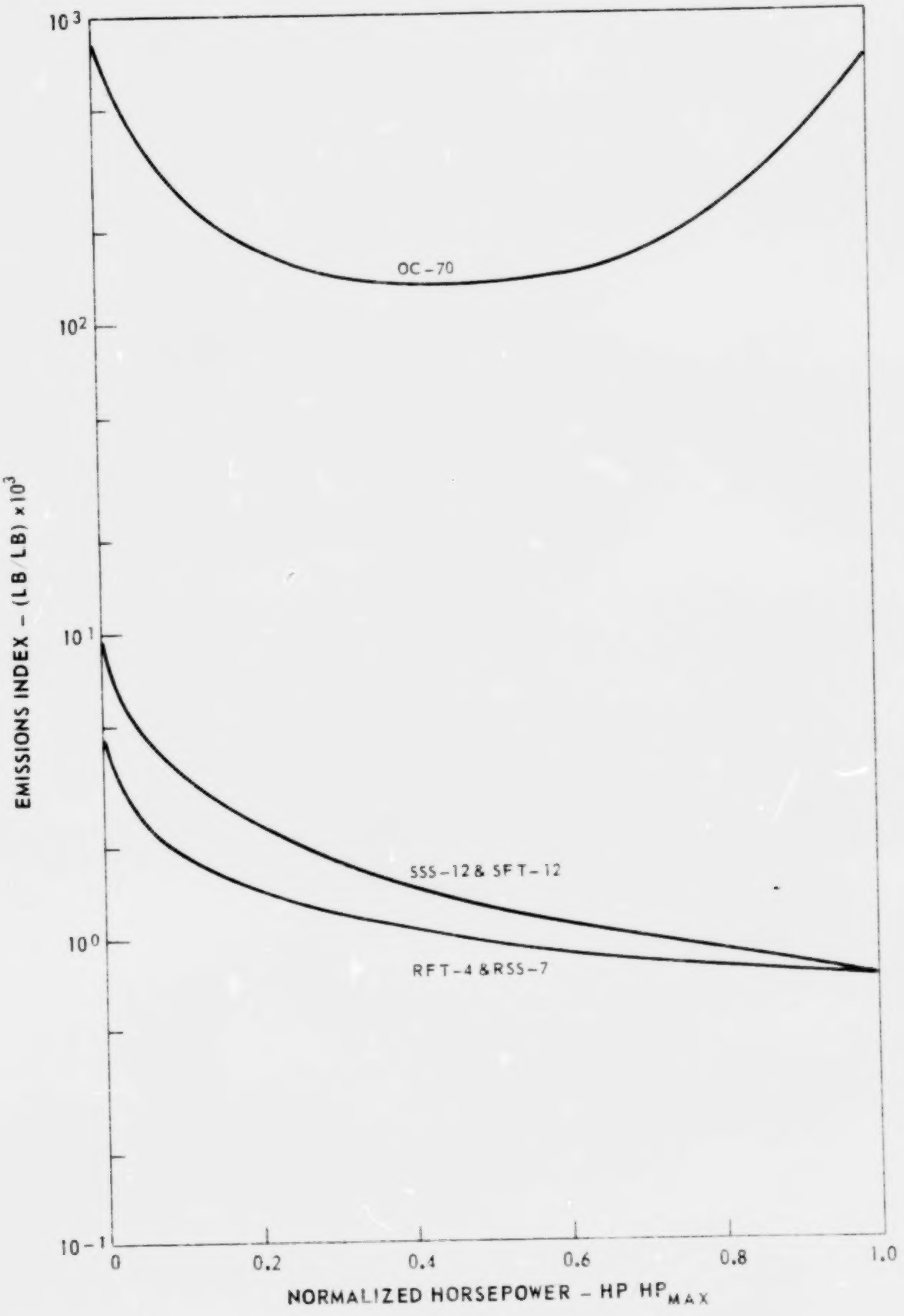
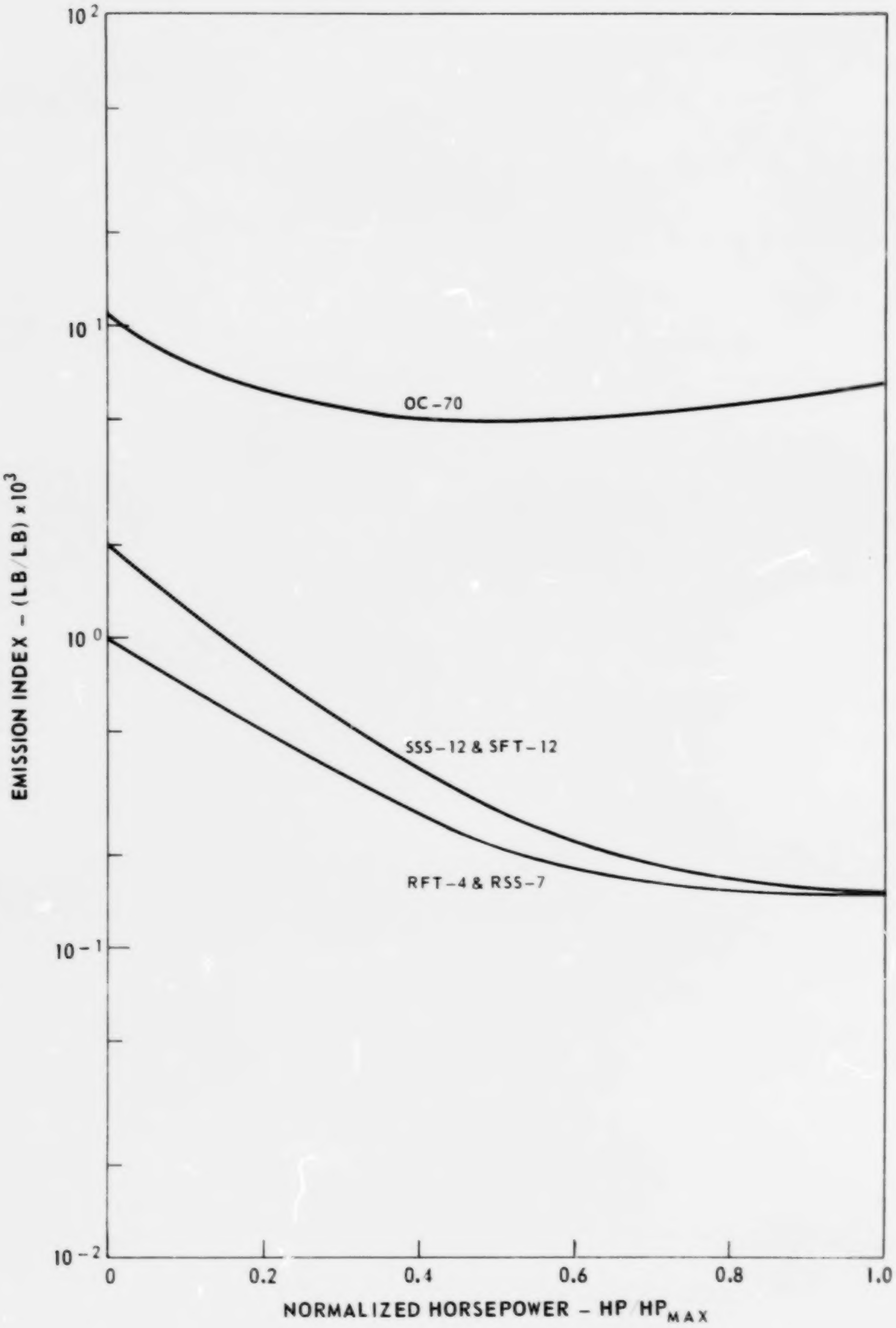


FIG. 32

EMISSIONS INDEX VERSUS NORMALIZED HORSEPOWER
UNBURNED HYDROCARBONS (UHC) (EXPRESSED AS C_6H_{14})



Although the emissions data base shown in Figs. 30, 31, and 32 is believed to be a best estimate based on current data, more complete test data, collected and correlated on a common basis by several contractors, will be available for use in predicting the operating characteristics of combustors which will have been designed from their inception as low-emission devices for the automobile application upon the completion of current combustor development studies being sponsored by OAP.

The emissions index was chosen for use in computing pollutant emissions because it relates pollutant production directly to fuel flow by a constant factor. Emissions are also commonly given in parts per million (ppm) of exhaust by volume. The EI is related to ppm by the following equation (Ref. 18):

$$EI = \frac{ppm}{1000} \left(\frac{MW_s}{MW_e}\right) (1 + AF),$$

where AF = air/fuel ratio (lb of air per lb of fuel)

MW_e = molecular weignt of exhaust

MW_s = molecular weight of species of pollutant

In Fig. 30, the emissions indices for the oxides of nitrogen (NO_x) are expressed as nitrogen dioxide (NO_2, $MW_s = 46$), and in Fig. 32 the emissions indices for UHC are expressed in terms of hexane (C_6H_{14}, $MW_s = 86$).

The computer program calculates all emission (and fuel consumption) data on the basis of steady-state input curves, with no consideration of the variations in EI or sfc which might occur under transient conditions, since there is a lack of suitable data. Similarly, cold start/hot start differentiation, which critically affects the ability of the Otto-cycle engine to meet UHC and CO emissions standards, has not been accounted for in the analysis. In both cases, neglect of these factors is expected to have lesser effects on the gas turbine engines studied because of their overall lean operation during acceleration and deceleration, and because of the relatively smaller cold quenching surfaces within the combustion chamber, although the magnitude is unknown at this time.

Performance Criteria

The vehicle performance criteria are based mainly on the OAP "Vehicle Design Goals - Six-Passenger Automobile" (Ref. 11) but also include performance requirements selected by UARL. These criteria, which are detailed below, include fuel economy and emissions over three driving cycles, acceleration and grade speed requirements, and constant-speed fuel economy.

Fuel density was assumed to be 7.0 lb/gal for the gas turbine-powered vehicles and 6.2 lb/gal (gasoline) for the OC-70 engine. Lower heating values (LHVs) for the turbine fuel and gasoline were taken as 18,400 and 18,650 Btu/lb, respectively.

Constant-Speed Fuel Economy

A fuel economy figure, in mpg and Btu/mi, is computed for the vehicles operating at constant speed, on a level road, at speeds of 20, 30, 40, 50, 60, 70, and 80 mph. Vehicle weight is W_{test}.

Driving Cycles

As enumerated earlier, three driving cycles were selected to measure vehicle performance. One is the urban driving cycle which appears in the July 15, 1970 Federal Register (the Federal Driving Cycle, or FDC). The speed profile for this 1370-sec cycle is shown in Fig. 23, along with the power required to operate the 3850-lb SSS-12-powered vehicle at the specified speed levels. Average power for the cycle is 10.6 hp including 1.3 accessory hp*, and average speed is 19.6 mph over the 7.5-mi cycle distance.

A heavy-duty driving cycle was developed at UARL to simulate suburban-type driving, which includes high-speed freeway operation. This 1080-sec cycle covers 14.7 mi and requires an average of 34.3 hp for a 3850-lb vehicle. The UARL cycle includes grades up to 2.0% and several bursts of high acceleration to simulate entrance and merging onto a highway. The average speed over the cycle is 48.9 mph, compared with 19.6 mph for the Federal Register cycle. The speed and grade profiles, and a horsepower profile for this cycle, are shown in Fig. 24. The UARL heavy-duty cycle was chosen to be representative of heavy-duty suburban and freeway driving in order to provide engine life design inputs, and no attempt was made to optimize it to any set of criteria other than the authors' judgment.

The third driving cycle (UARL typical driving cycle) was used primarily to measure vehicle fuel economy under country driving conditions, and is shown in Fig. 25. This 1154-sec cycle is believed to be representative of typical country or suburban driving, but as in the previous case, represents the authors' judgment as to what comprises a typical cycle. It requires 29.3 hp for the 3850-lb SSS-12-powered car, including 1.3 accessory hp. Average speed is 47.6 mph over the 15.3-mi cycle.

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* Average power for the cycle is 9.3 hp, if the accessory power is zero.

Vehicle weight is taken as W_{test}, and all performance is on a level grade except where grades are specified in the UARL cycles. The maximum allowable UHC, CO, and NO_x emissions specified for vehicles operating over the FDC* are:

Pollutant	Grams per Mile (maximum)	
	OAP Research Goals (Ref. 11)**	1976 Federal Standards (July 2, 1971 Federal Register)
Hydrocarbons	0.14	0.41
Carbon Monoxide	4.70	3.40
Oxides of Nitrogen	0.40	0.40

where the oxides of nitrogen (NO_x) are measured or computer as NO_2, and hydrocarbons are computed as hexane (C_6H_{14}).

Acceleration and Grade Speed Requirements

Six performance regimes comprise the acceleration and grade speed criteria. They are taken from the OAP vehicle design goals and are defined as follows:

1. Acceleration from a standing start: The minimum distance to be covered in 10.0 sec is 440 ft. Acceleration is on a level grade and vehicle weight is W_{test} (curb weight plus 300 lb).

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* The most recent version of the OAP vehicle design goals (Revision "B") specifies that the vehicles shall be tested for emissions using the driving cycle which appears in the November 10, 1970 Federal Register. An earlier version of the design goals specified an almost identical cycle which appeared in the July 15, 1970 Federal Register. Revision "B" was not received by UARL until after the completion of the computer program for this report so that the results herein apply to the July 15, 1970 driving cycle.

** These goals are currently being revised to reflect the cold start weighting procedure identified in the July 2, 1971 Federal Register.

2. Acceleration from a standing start: The maximum time to reach a speed of 60 mph is 13.5 sec. Acceleration is on a level grade and vehicle weight is W_{test}.

3. Acceleration in merging traffic: The maximum time to accelerate from a constant speed of 25 mph to a speed of 70 mph is 15.0 sec. Acceleration is on a level grade and vehicle weight is W_{test}.

4. Acceleration, DOT High-Speed Pass Maneuver*: The maximum time and maximum distance to go from an initial speed of 50 mph with the front of the vehicle 100 ft behind the back of a 55-ft truck traveling at a constant 50 mph to a position where the back of the vehicle is 100 ft in front of the front of the 55-ft truck is 15 sec and 1400 ft. The entire maneuver takes place in a traffic lane adjacent to the lane in which the truck is operated. The vehicle speed during the maneuver cannot exceed 80 mph. Vehicle weight is W_{test} and acceleration is on a level grade.

5. 5% Grade Speed: The vehicle must be capable of achieving a speed of 70 mph up a 5% grade and maintaining this speed for a period of 100 seconds when preceded and followed by continuous operation at 60 mph on the same grade.

6. 30% Grade Speed: The vehicle must be capable of starting from rest on a 30% grade and accelerating to 15 mph and maintaining it.

Maximum vehicle acceleration is commanded for these six performance requirements to determine if they can be met or exceeded. Results are given in terms of the minimum (or maximum) value of each performance criterion; for example, the minimum time in which the vehicle can accelerate from a standstill to 60 mph.

A summary of the vehicle performance criteria and a listing of the results which are reported for each are given in the following table. Emissions data are reported for CO, UHC, and NO_x.

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* The DOT High-Speed Pass Maneuver is detailed in Appendix III on page 181.

Vehicle Performance Criteria	Results
Constant-Speed Fuel Economy	Fuel Economy (mpg), (Btu/mi)
Federal Register Driving Cycle (FDC)	Fuel Economy (mpg), Emissions (gm/mi)
UARL Heavy-Duty Driving Cycle	Fuel Economy (mpg)
Typical Suburban Driving Cycle	Fuel Economy (mpg)
Maximum Distance covered in 10.0 sec from standstill	Distance (ft)
Minimum Time for 0-60 mph	Time (sec)
Minimum Time for 25-70 mph	Time (sec)
DOT High-Speed Pass Maneuver	Time (sec), Distance (ft)
5% Grade Speed	Speed (mph)
30% Grade Speed	Speed (mph)

Discussion of Results

The technique used to obtain results for the vehicle performance analysis was to run computer simulations for all performance criteria with all the base-line vehicles to obtain a set of base-line results for each case. Then, the effects of variations of individual vehicle parameters on the base-line results were obtained by rerunning all the computer simulations and holding all vehicle parameters fixed at their base-line values except the one whose variation was to be studied. These sensitivity analyses were made to determine the effects of variations of accessory horsepower, vehicle weight, fuel consumption (sfc), and maximum engine horsepower.

For convenience in discussing the vehicle performance data, it should be noted that a complete description of the base-line vehicles can be found in Table III (P. 57) which contains the engine and transmission combinations, and base-line vehicle weights, and Table IV (page 70), which summarizes base-line vehicle parameters such as frontal area, drag coefficient, and accessory power loads. All base-line vehicle engines have 150 installed horsepower.

A better perspective of the formal results of the vehicle performance analysis can be gained by first examining the curves of Figs. 33 and 34. Figure 33 illustrates a breakdown of the constant-speed power requirements as a function of speed for the base-line vehicle with the simple-cycle, single-shaft gas turbine engine. The power requirements for the other base-line vehicles are quite similar, the only differences being in rolling resistance (because of the different vehicle weights) and variations in the transmission losses. For example, the power requirement for the vehicle with the OC-70 engine, which, at 4300 lb, is the heaviest vehicle, is 41.5 hp at 60 mph, compared with 38.0 hp for the SSS-12 vehicle (3850 lb). Figure 34 shows the top speed and 0-to-60 mph acceleration time for the base-line vehicle with the SSS-12 engine, as a function of maximum engine horsepower. For the base-line engine size of 150 hp, the top speed is 105 mph and the 0-to-60 mph acceleration time is 10.2 sec.

Constant-Speed Fuel Economy

The constant-speed fuel economy (in mpg and Btu/mi) of the base-line vehicles, as a function of speed, is shown in Table VI. The fuel economy, in mpg, is also shown graphically in Fig. 35. For this computation, the vehicles are operated on a level road with weight equal to W_{test}. The regenerated engines are seen to deliver the best fuel economy over the range of speeds considered (20 to 80 mph), with the RSS-7 engine attaining a maximum of 20.2 mpg at 60 mph. The simple-cycle engines have a poorer overall fuel economy than the other engines, but it does not drop off as rapidly at the higher speeds; they have a better fuel economy than the OC-70 engine at speeds above 65 mph.

The sensitivity of the constant-speed fuel economy of the four engines to accessory horsepower, vehicle weight, sfc, and maximum engine horsepower is shown in Figs. 36, 37, and 38 for speeds of 30, 50, and 70 mph, respectively.

The vehicle weight variations (± 15%) are taken about the base-line value of W_{test}. The variations in fuel economy due to vehicle weight are caused by changes in the rolling and inertial resistances of the vehicle and, in a secondary way, to the corresponding changes in transmission losses. The sfc variations were obtained by increasing or decreasing the sfc values at all normalized powers by a constant percentage. A fuel economy figure proportional to the reciprocal of mpg (such as Btu/mi) would plot as a straight line versus sfc variation. The effect, on fuel economy, of varying maximum (rated) engine power is shown in the lower right-hand set of curves. Since the power at any speed is constant, a decrease in maximum

CONSTANT SPEED POWER REQUIREMENTS VERSUS SPEED

VEHICLE WEIGHT = 3850 LB

OAP ROLLING RESISTANCE FORMULA

FRONTAL AREA = 25 FT^2

DRAG COEFFICIENT = 0.48

ACCESSORY POWER = 4.0 HP

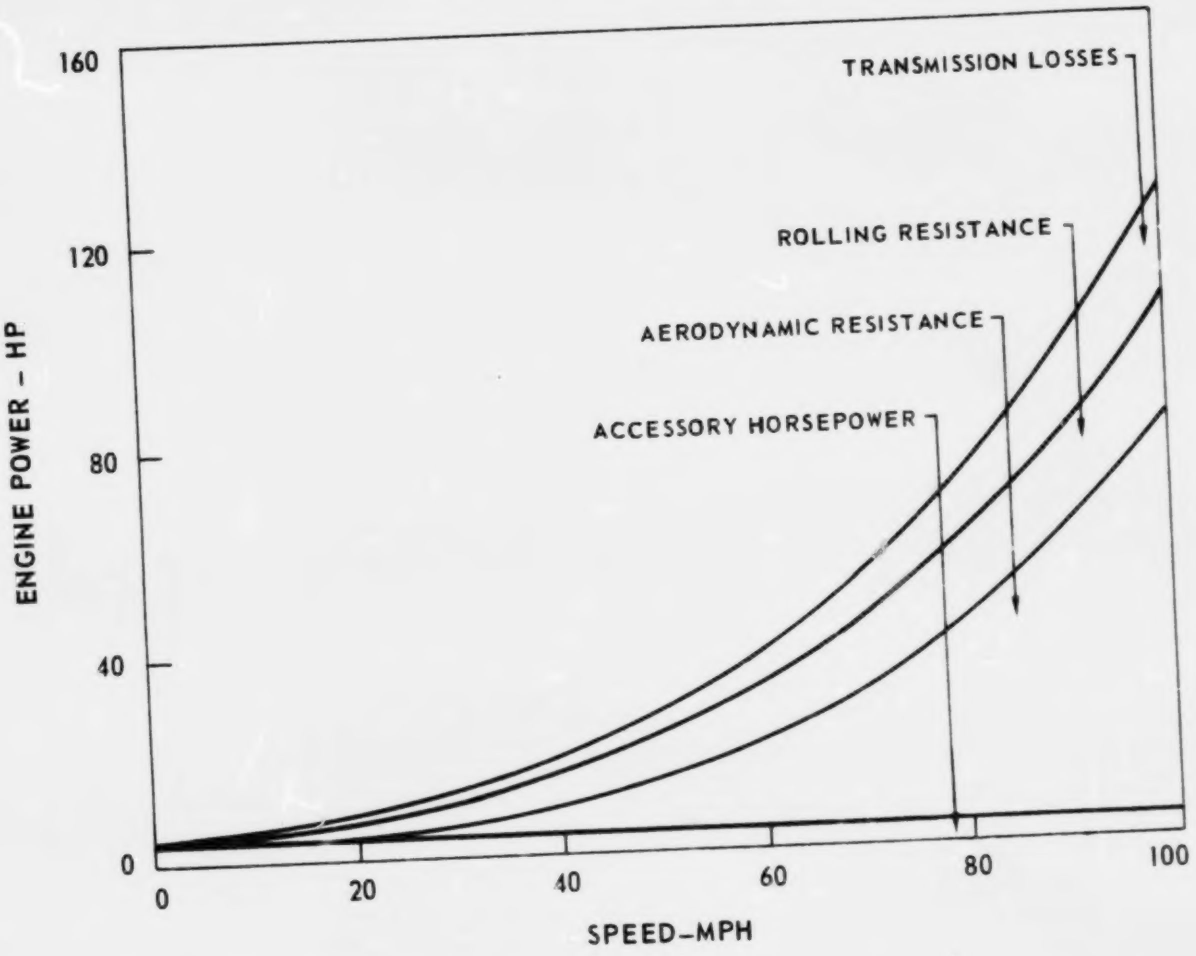


FIG. 34

TOP SPEED AND ACCELERATION TIME VERSUS MAXIMUM ENGINE POWER

VEHICLE WEIGHT – 3850 LB

OAP ROLLING RESISTANCE FORMULA

FRONTAL AREA – 25 FT^2

DRAG COEFFICIENT – 0.48

ACCESSORY POWER – 4.0 HP

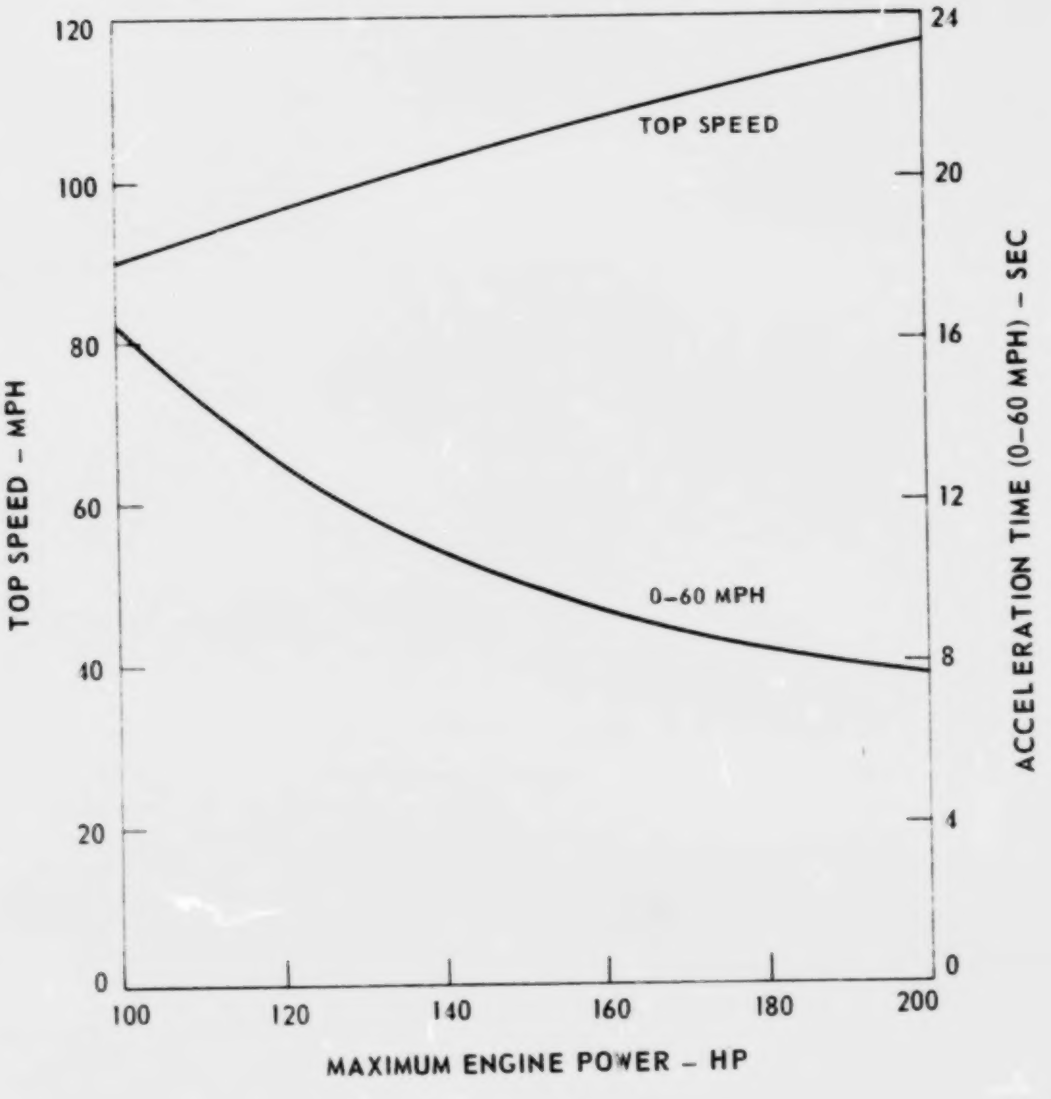


TABLE VI

FUEL ECONOMY, CONSTANT-SPEED
BASE LINE VEHICLES

mpg (Btu/mile)

Ambient Temperature = 60 F
Accessory Power = 4.0 hp

Engine \ Speed, mph	20	30	40	50	60	70	80
RFT-4	10.16 (12,682)	13.31 (9,676)	15.87 (8,155)	16.77 (7,683)	16.12 (7,990)	14.39 (8,949)	12.40 (10,389)
SSS-12	7.98 (16,149)	10.03 (12,842)	11.89 (10,829)	12.60 (10,222)	12.57 (10,243)	11.92 (10,806)	11.16 (11,537)
SFT-12	8.01 (16,086)	10.12 (12,733)	11.91 (10,817)	12.71 (10,132)	12.70 (10,138)	12.09 (10,654)	11.32 (11,374)
RSS-7	12.51 (10,299)	16.45 (7,828)	18.98 (6,785)	19.90 (6,473)	20.22 (6,370)	18.11 (7,111)	15.85 (8,125)
OC-70	10.65 (10,855)	12.62 (9,160)	14.10 (8,199)	13.71 (8,435)	12.91 (8,954)	11.48 (10,070)	9.89 (11,696)

FIG. 35

CONSTANT-SPEED FUEL ECONOMY-MPG

ACCESSORY POWER = 4.0 HP

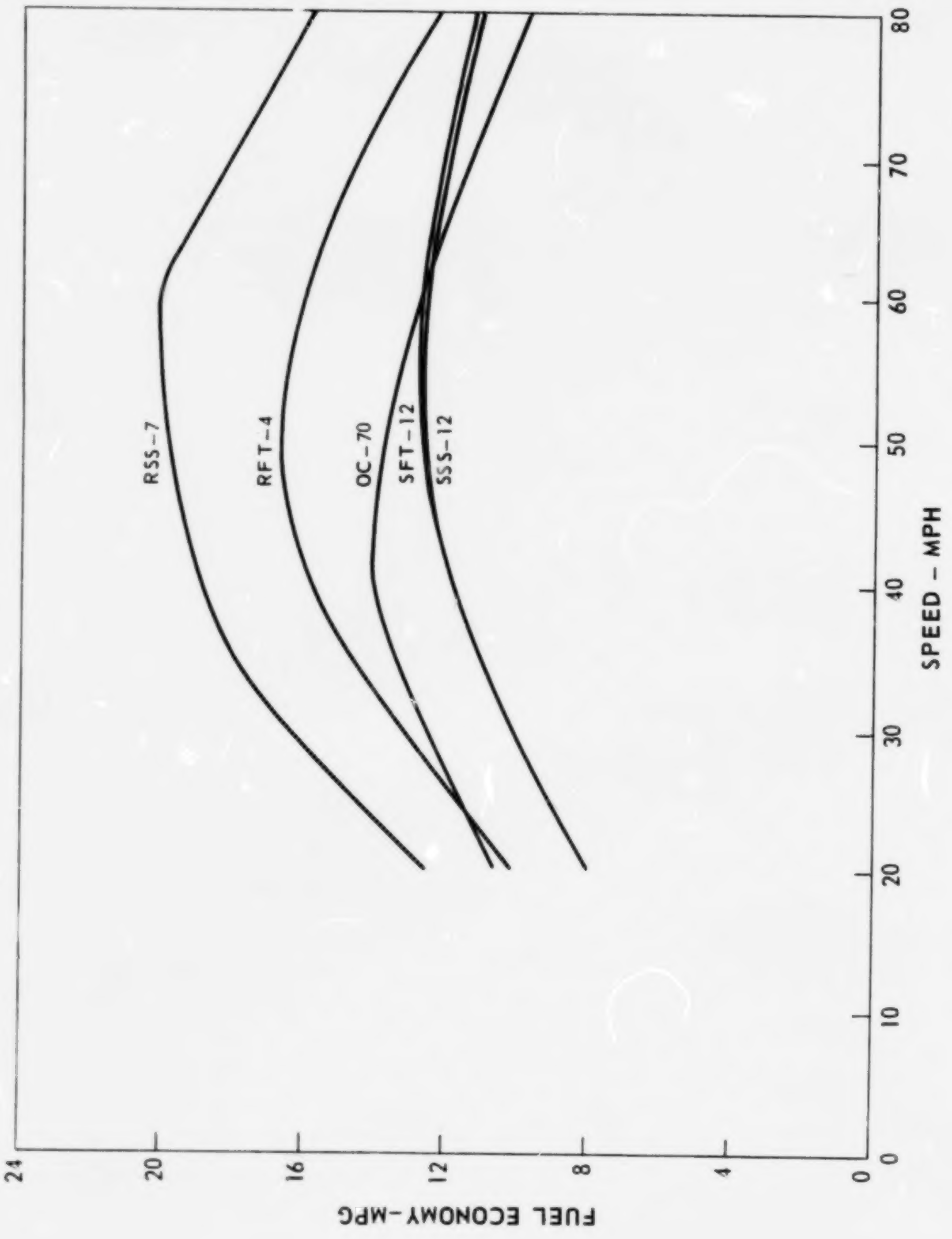


FIG. 36

FUEL ECONOMY AT CONSTANT 30-MPH SPEED

EFFECT OF VARIATION OF VEHICLE PARAMETERS ON FUEL ECONOMY

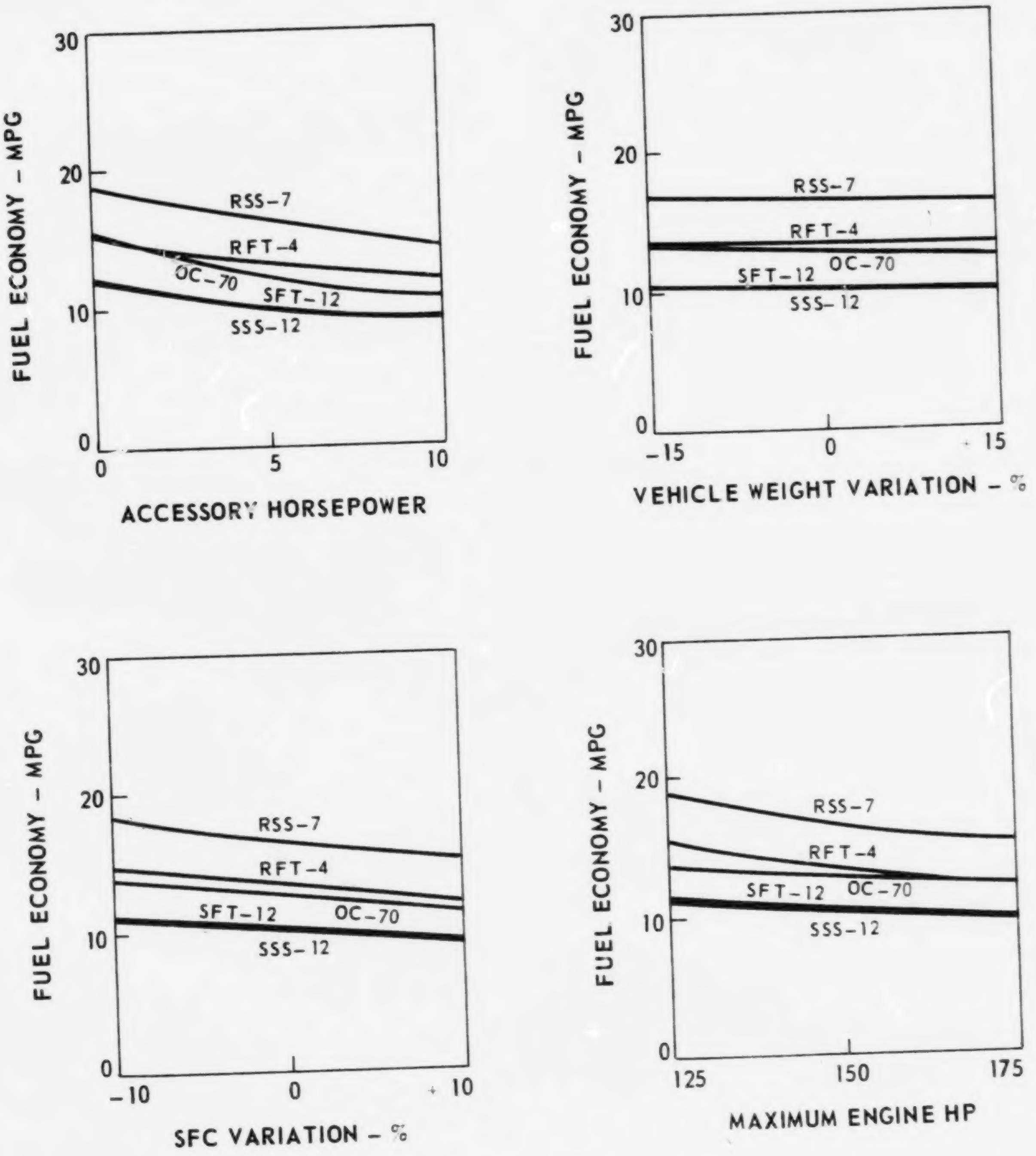
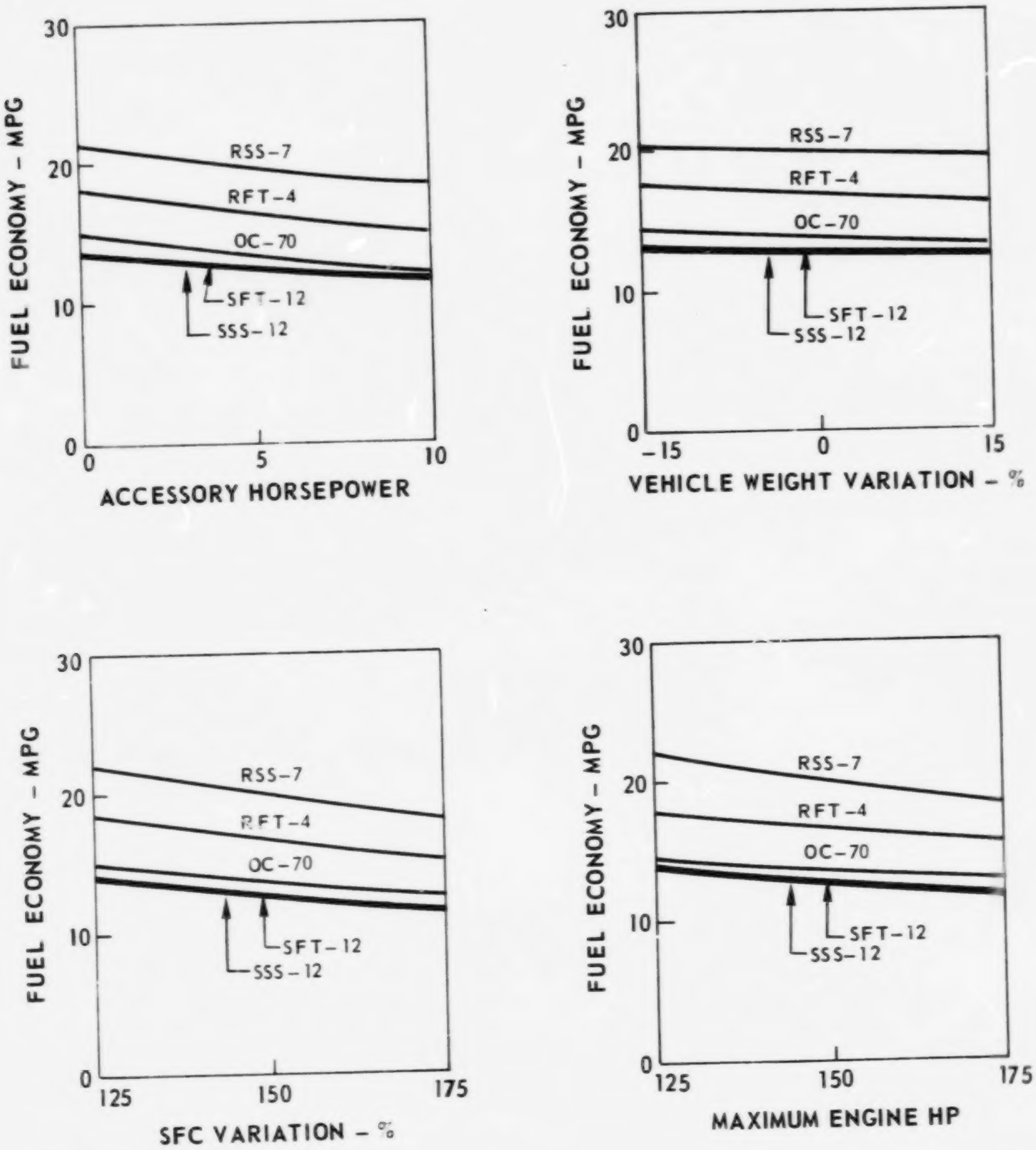


FIG. 37

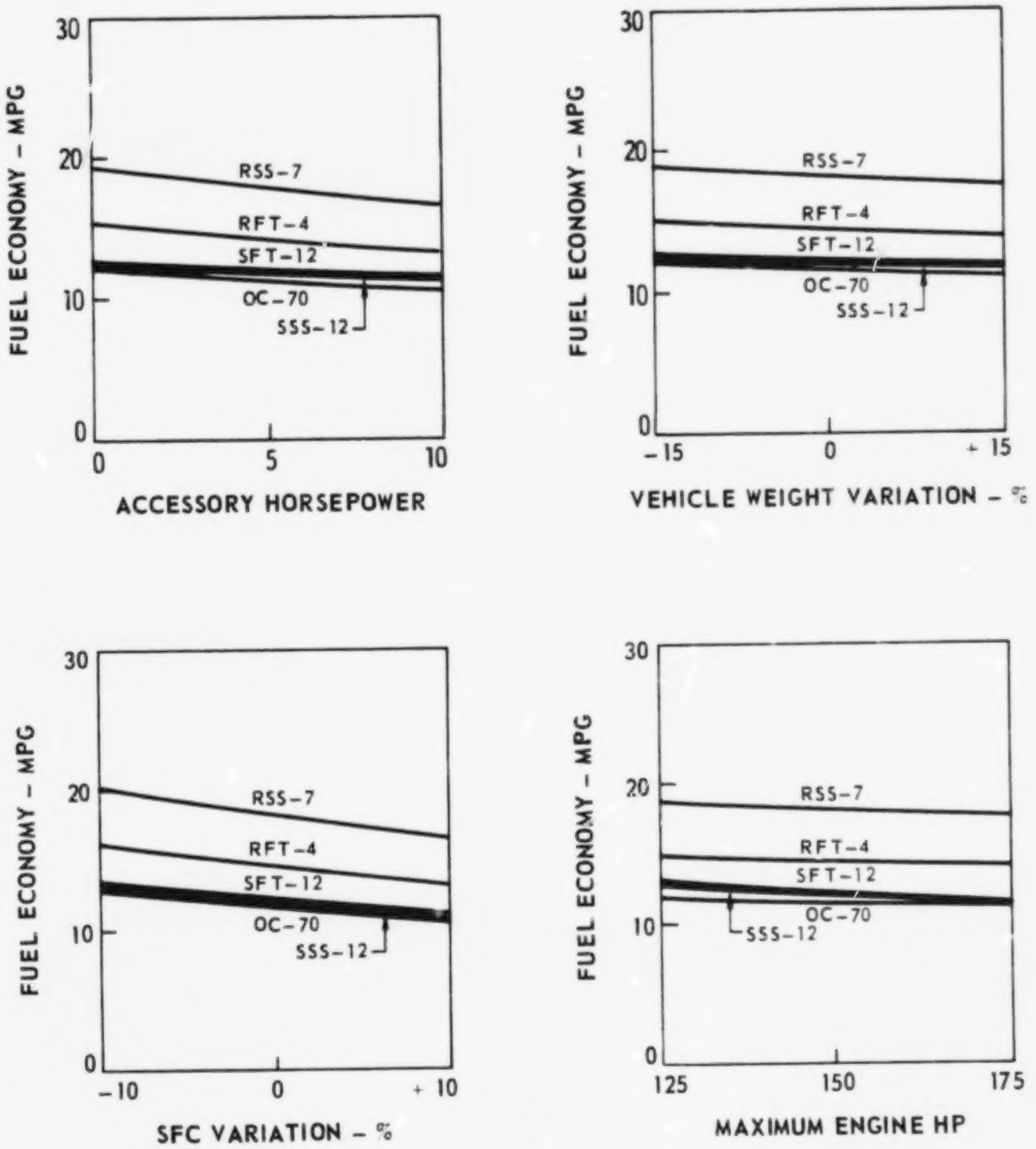
FUEL ECONOMY AT CONSTANT 50-MPH SPEED

EFFECT OF VARIATION OF VEHICLE PARAMETERS ON FUEL ECONOMY



FUEL ECONOMY AT CONSTANT 70-MPH SPEED

EFFECT OF VARIATION OF VEHICLE PARAMETERS ON FUEL ECONOMY



engine power corresponds to an increase in average normalized horsepower for the cycle, and it can be seen from the slope of the sfc curves of Fig. 28 that a decrease in fuel consumption (an increase in mpg) will result.

Driving Cycles

Figure 39 shows a fuel economy comparison among the base-line vehicles operated over the three selected driving cycles.

For the Federal Driving Cycle, the RSS-7 engine is seen to deliver the greatest fuel economy, with an average of 11.54 mpg, while the OC-70 engine delivers 9.49 mpg. The basic regenerated engine (RFT-4) is just below the OC-70 in fuel economy, with an average of 9.36 mpg, while the single-shaft engines have economy figures of 7.34 and 7.38 mpg for the SSS-12 and SFT-12, respectively.

The fuel economy over the UARL typical-duty cycle is better than that over the UARL heavy-duty cycle for all engines because, while the average speeds over the two cycles are about the same (~ 48 mph), the average power for the typical-duty cycle is about 5 hp less than for the heavy-duty cycle. The regenerated engines have the best fuel economy, as would be expected from Fig. 28, and the single-shaft and OC-70 engines deliver about the same fuel economy because of the higher average horsepowers required for these driving cycles.

These results are summarized in Table VII along with the average power requirements for each vehicle and cycle. In addition, Table VII shows the effects on fuel economy of using the infinitely variable hydromechanical transmission (HMT). The lower efficiency of this transmission is seen to produce a decrease in fuel economy of about 24% for the FDC, about 8% for the UARL typical-duty cycle, and about 5% for the UARL heavy-duty cycle. These decreasing percentages reflect the facts that each driving cycle, in the order mentioned, expends power at higher speeds than the cycle previous to it (for example, high-speed cruise as opposed to low-speed acceleration), and that the efficiency of the HMT is lower, relatively, at low vehicle speeds than that of the other three transmissions (Fig. 29).

The sensitivity of fuel economy to the same parameters as previously discussed for constant-speed conditions is shown in Fig. 40 for the Federal Driving Cycle.

The pollutant emissions, in gm/mi, for the base-line vehicles operating over the FDC are presented in Table VIII along with the current OAP design goals and 1976 Federal Emissions Standards. The simple-cycle engines are about 60% above the OAP design goal of 0.4 gm/mi of NO_x (oxides of nitrogen expressed as NO_2), with 0.65 and 0.64 gm/mi for the SSS-12 and SFT-12 engines, respectively. However, the estimates for the NO_x EI are considered quite conservative since, as described earlier, the NO_x EI data were based on measurements taken from the UACL PT6A-27 engine, and NO_x emissions were not a consideration at the time the combustors were designed for this engine.

FIG. 39

FUEL ECONOMY FOR VARIOUS DRIVING CYCLES

ACCESSORY POWER = 1.3 HP

AMBIENT TEMPERATURE = 60°F

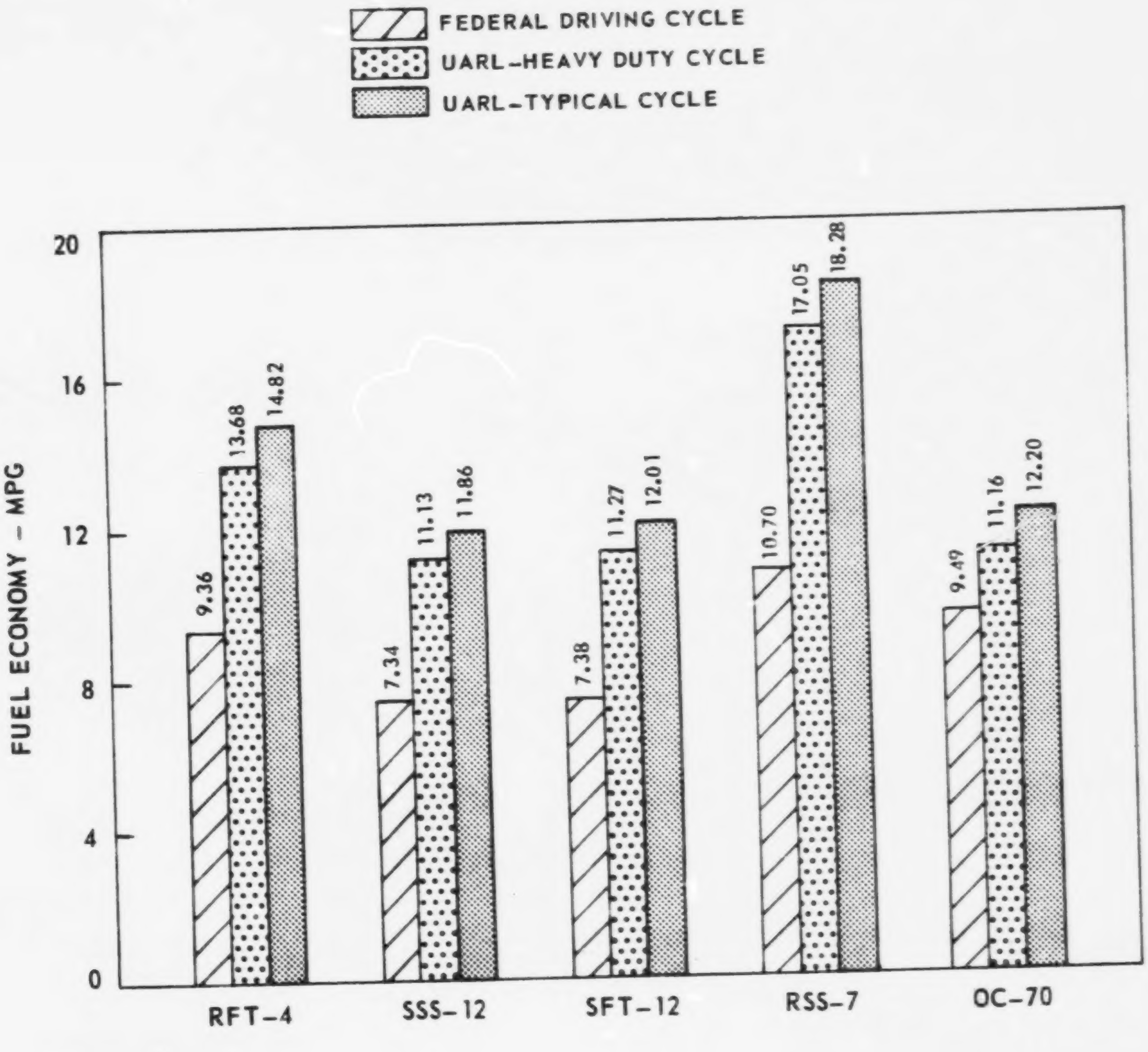


TABLE VII

FUEL ECONOMY, VARIOUS DRIVING CYCLES

Accessory Power = 1.3 hp
Ambient Temperature = 60 F

		FDC		UARL Heavy Duty		UARL Typical	
Engine	Transmission	Average Power	MPG	Average Power	MPG	Average Power	MPG
SSS-12	8-Speed	10.60	7.34	34.29	11.13	29.31	11.36
SSS-12	HMT	13.10	6.87	36.78	10.56	31.19	11.17
SFT-12	4-Speed	10.40	7.38	33.51	11.27	28.65	12.01
RSS-7	8-Speed	11.54	10.70	34.55	17.05	29.53	18.28
RFT-4	4-Speed	10.84	9.36	34.5[illegible]	13.68	29.55	14.82
OC-70	3-Speed	11.67	9.49	37.21	11.16	31.77	12.20

FIG. 40

VEHICLE PERFORMANCE – FEDERAL DRIVING CYCLE

EFFECT OF VARIATION OF VEHICLE PARAMETERS ON FUEL ECONOMY

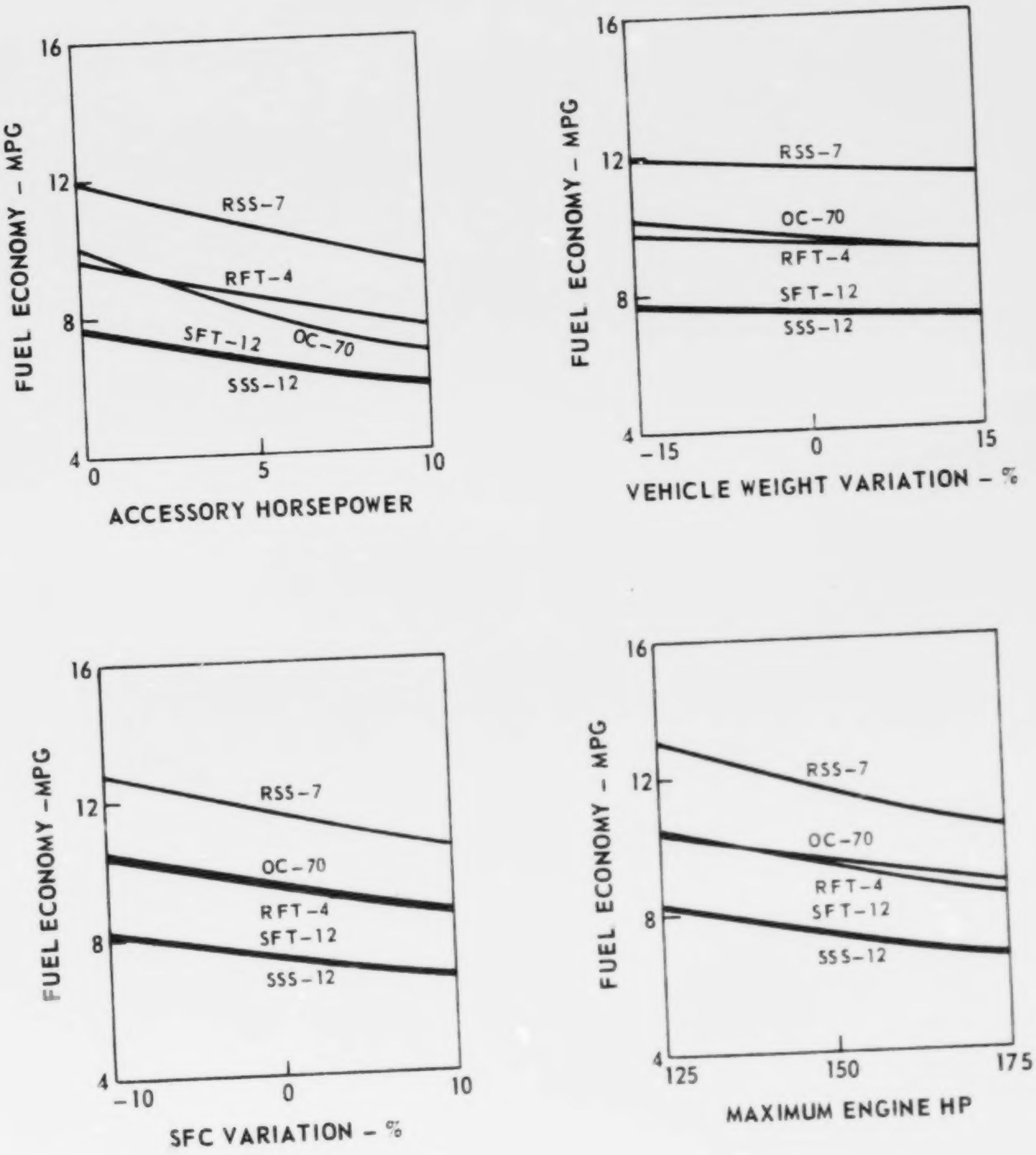


TABLE VIII

EMISSIONS, FEDERAL DRIVING CYCLE
BASE-LINE VEHICLES

Ambient Temperature = 60 F
Accessory Power = 1.3 hp

Engine	NO_x - gm/mi	CO - gm/mi	UHC - gm/mi
RFT-4	2.89	0.88	0.26
SSS-12	0.65	2.11	0.61
SFT-12	0.64	2.11	0.61
RSS-7	2.33	0.71	0.21
OC-70	4.16	117.0	2.33
OAP Research Goals (Revision B, February 11, 1971)	0.4	4.7	0.14
1976 Federal Emissions Standards (using 1975-76 procedure as specified in July 2, 1971 Federal Register)	0.4	3.4	0.41

Reductions in NO_x emissions of from 37% (Ref. 21) to 67% (Ref. 19) have been reported for gas turbines where design efforts have been made to decrease NO_x emissions; consequently, the authors believe that similar improvements in simple-cycle engines with combustors designed for low emissions could meet or improve on the OAP 0.4 gm/mi NO_x design goal. The regenerated engines are estimated to produce 2.9 and 2.3 gm/mi of NO_x for the RFT-4 and RSS-7 engines, respectively, which are slightly higher than the 1.9 gm/mi estimated in Ref. 21. These levels of NO_x are 3 to 4 times those of the single-shaft engines and may be attributed to the higher combustor inlet temperatures of the regenerated engines. The untreated OC-70 engine is estimated to produce 4.2 gm/mi of NO_x in comparison with the 5 to 6 gm/mi currently being measured at OAP facilities.

It is seen that all of the gas turbine engines have emission levels of CO well below the OAP design goal maximum of 4.7 gm/mi, with the simple-cycle engines below the goal by about 55% and the regenerated engines by at least 80%. The untreated OC-70 engine produces 117 gm/mi of CO, thereby exceeding the permissible maximum by a factor of 25.

None of the study engines meet the OAP design goals for UHC. The regenerated engines produce the least UHC with levels of 0.26 and 0.21 gm/mi for the RFT-4 and RSS-7 engines, respectively. Both simple-cycle engines produce 0.6 gm/mi of UHC, while the OC-70 engine produces 2.3 gm/mi.

The apparent failure of the gas turbines to meet UHC goals may disappear when estimates derived from the testing of combustors designed low emissions replace the estimates made for this study. In any event, it is obvious that little, if any, increases in CO and UHC emission values can be accepted as the price for designing low-NO_x combustors.

Figures 41, 42, and 43 illustrate the effects of variations of accessory horsepower, vehicle weight, sfc, and maximum engine horsepower on the emissions of NO_x, CO, and UHC, respectively. (Note the use of the folded scale for the CO emissions of the OC-70 engine.) Generally, the emissions levels were insensitive to the vehicle weight variations of ± 15%, except for the NO_x emissions of the OC-70 engine. Since the pollutant emissions (gm/mi) for the cycle are proportional to the amount of fuel used, which is proportional to sfc, the variations of NO_x, CO, and UHC with sfc are linear.

Acceleration and Grade Speed Requirements

As discussed earlier, all vehicle transmission characteristics are represented by single-valued functions of efficiency versus percent output speed in the automobile performance computer programs. This method, while believed to produce meaningful estimates of fuel economy and emissions, can lead to erroneous results when

FIG. 41

K-971017-4

VEHICLE PERFORMANCE – FEDERAL DRIVING CYCLE

EFFECT OF VARIATION OF VEHICLE PARAMETERS ON NO_x EMISSIONS

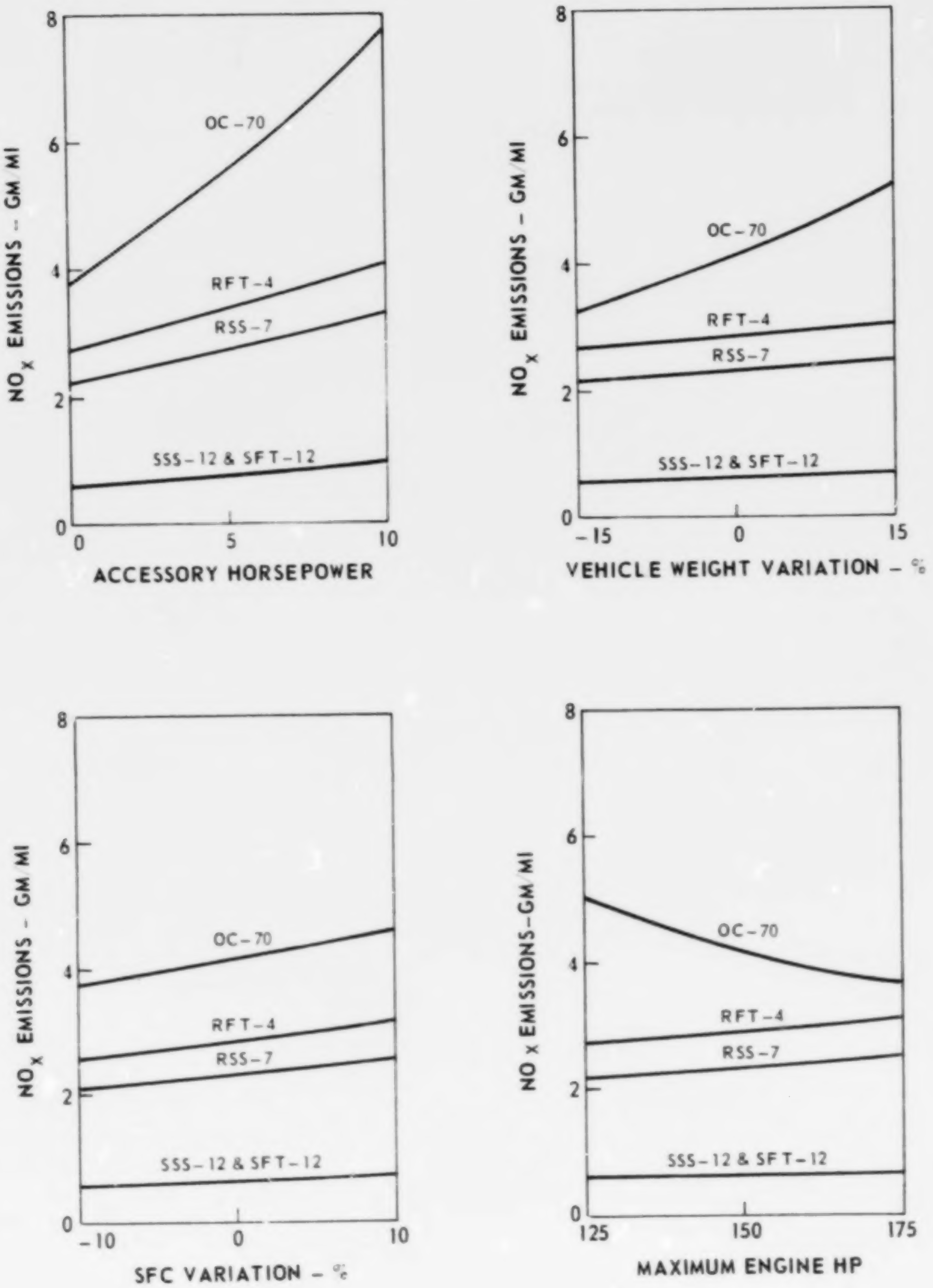


FIG. 42

VEHICLE PERFORMANCE FEDERAL DRIVING CYCLE

EFFECT OF VARIATION OF VEHICLE PARAMETERS ON CO EMISSIONS

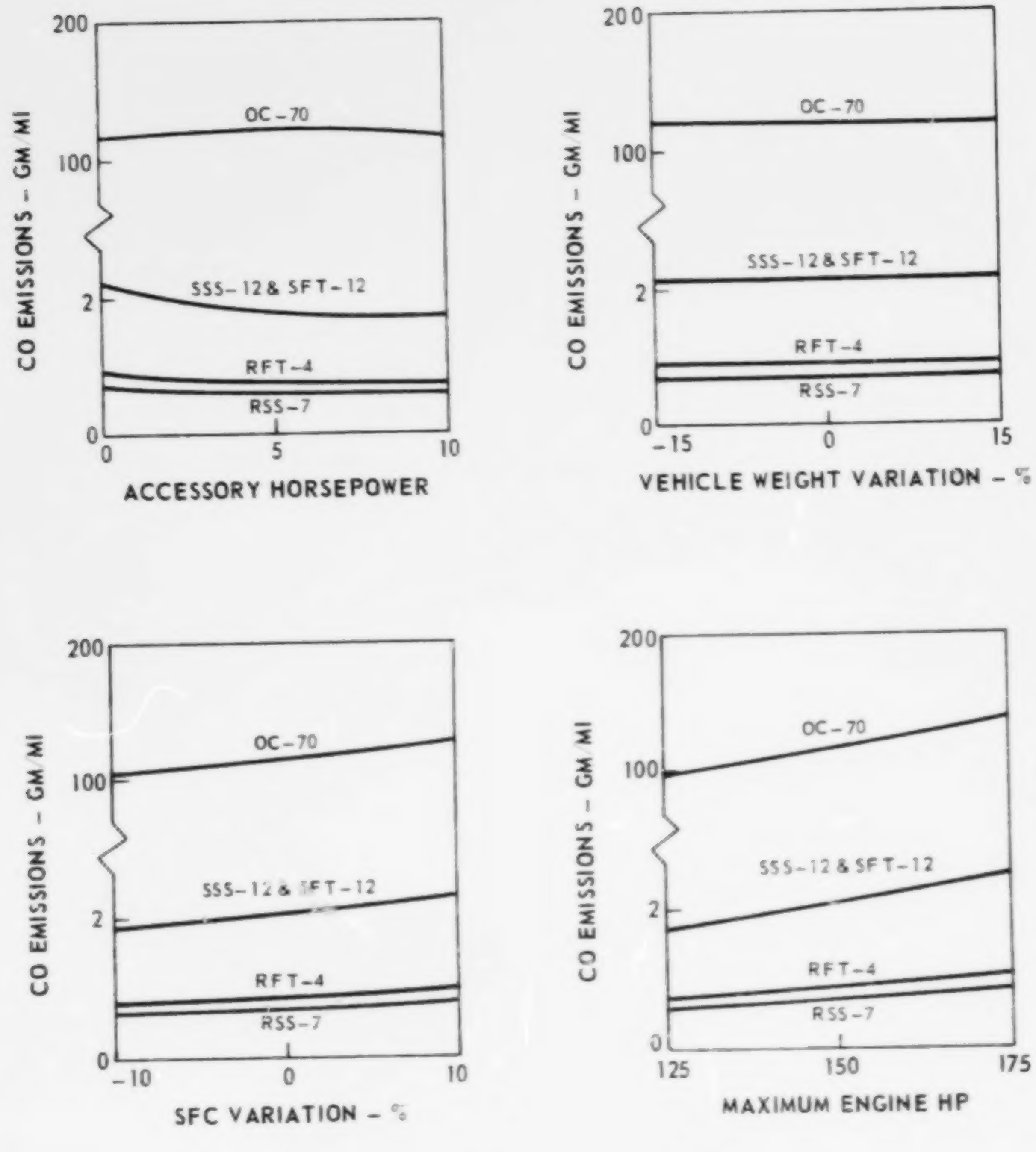
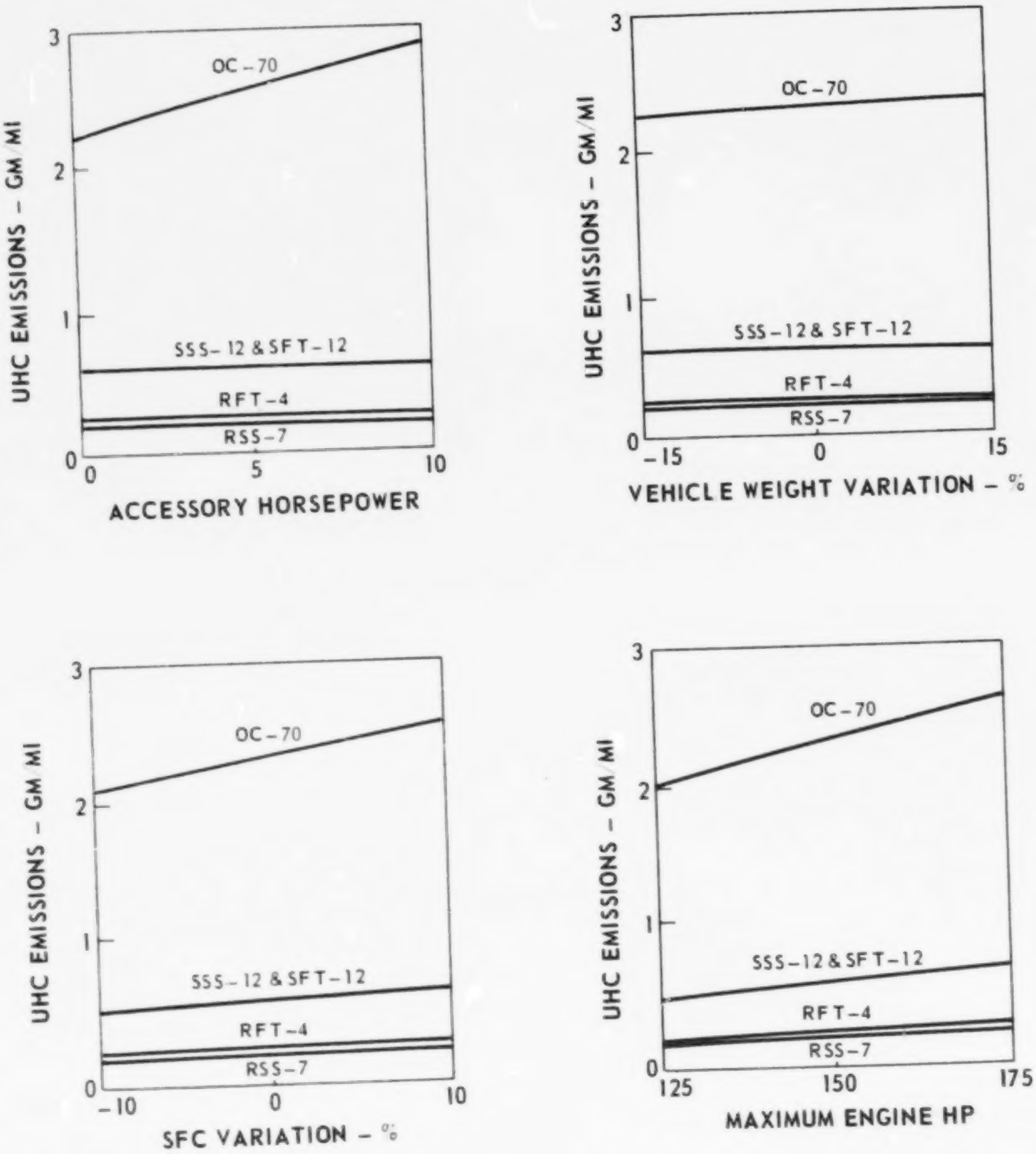


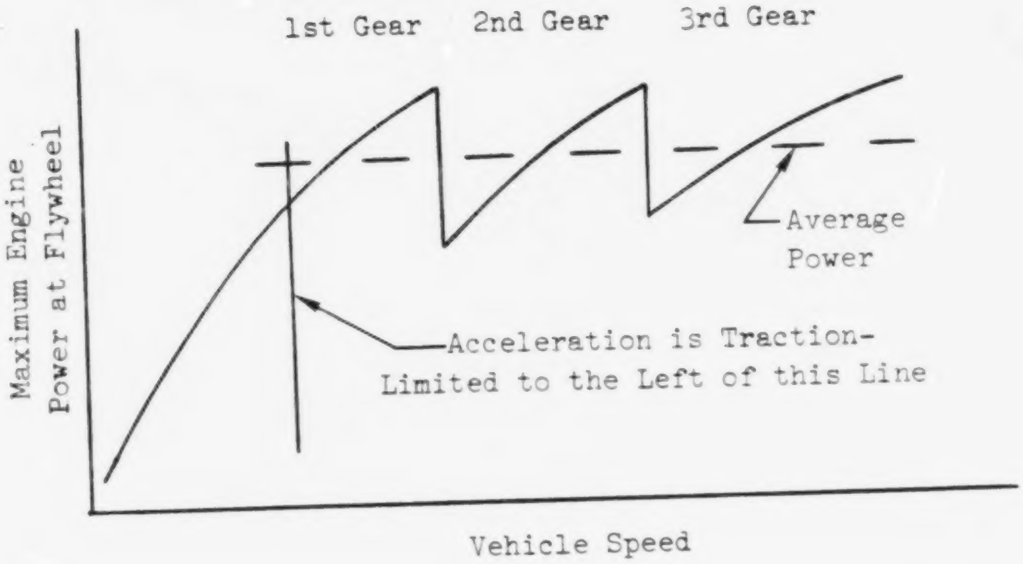
FIG. 43

VEHICLE PERFORMANCE – FEDERAL DRIVING CYCLE

EFFECT OF VARIATION OF VEHICLE PARAMETERS ON UHC EMISSIONS



determining acceleration performance under full power. The situation that occurs is illustrated in the following sketch for a typical vehicle with a three-speed transmission.



In the usual stepped transmission, the engine is geared (directly or indirectly) to the wheels, and maximum engine speed, hence maximum power, is developed only at certain vehicle speeds. The sketch, illustrating a full-throttle acceleration, shows how engine power increases with vehicle speed in each gear until an upshift is made, presumably at maximum engine speed and power. The average engine power over the acceleration run is clearly less than the maximum power available, and it is the average power which must be used in the vehicle performance simulation to obtain realistic performance results.

For the acceleration performance runs in this study, average engine power was obtained by using the appropriate engine power-speed curve and transmission gear ratios to create plots similar to the preceding sketch. In each of the study engines, the average power for each engine (the dashed line in the sketch) was then estimated, assuming that below about 25 mph the vehicle would be traction (rather than horsepower) limited. In other words, the average power estimates were based on the portions of the power-speed curves above 25 mph. The average power so estimated was used for the following performance computations: (1) 10-sec acceleration distance, (2) 0-60 mph acceleration time, (3) 25-70 mph acceleration time, and (4) the DOT High-Speed Pass Maneuver (HSPM). Full engine power was used on the 5% and 30% grade speed requirements because of gearing considerations.

The average power obtained by this technique for each of the base-line study engines equipped with the transmissions specified in Table IV is given in the following table, assuming a maximum flywheel output power of 150 hp.

Engine	Average hp	Transmission Type
RFT-4	143	4-speed
SSS-12	125	8-speed
SFT-12	143	4-speed
RSS-7	125	8-speed
OC-70	125	3-speed

For results involving variations of rated power, a new average power was determined from the base-line average by a simple ratio. (For example, the average power for a 175-hp OC-70 engine would be 125 x (175/150) = 146 hp.)

The results for the acceleration and grade speed requirements are presented in bar-chart form in Figs. 44 through 50, with data given for the base-line vehicle and for variations in vehicle weight and maximum engine horsepower. Where the performance criteria involved a minimum vehicle acceleration level, maximum vehicle performance was commanded to determine by what amount, if any, the vehicle could exceed the design goal.

Figure 44 shows the maximum distances which can be covered in 10 sec from a standstill, on a level grade, with vehicle weight equal to W_{test}. The minimum OAP goal is 440 ft, and all vehicles exceed this distance by at least 56 ft, the minimum improvement being that for the OC-70 engine with 125 hp.

Figure 45 shows the minimum 0-to-60 mph acceleration time for all vehicles. Acceleration is on a level grade and vehicle weight is W_{test}. The OAP design goal is 13.5 sec, which is met by all vehicles and all variations, except the 125-hp OC-70 engine. The free-turbine engined vehicles accelerate the most rapidly; e.g., the base-line vehicle with the SFT-12 engine (3840 lb) accelerates from 0 to 60 mph in 8.5 sec, while the OC-70 engined vehicle (4300 lb) required 11.3 sec. This result is not surprising, however, because the torque characteristic of the free turbine provides a higher average horsepower (table above) over the vehicle's speed range, and hence, greater acceleration.

Figure 46 shows the minimum time for a 25-to-70 mph acceleration on a level grade with vehicle weight equal to W_{test}. The OAP design goal for this acceleration regime is 15 sec. All base-line gas turbine vehicles perform the maneuver in less than 12.5 sec, while the 150-hp OC-70 vehicle requires 14.1 sec.

The results of the simulated DOT high-speed pass maneuver are shown in Figs. 47 (passing time) and 48 (passing distance). Vehicle weight is W_{test} and acceleration is on a level grade. The OAP goals for this maneuver are 15.0 sec and 1400 ft for total passing time and distance, respectively. These criteria are easily within the capabilities of all vehicles in all variations, except the 125-hp OC-70 engines which required 15.4 sec and 1399 ft for the maneuver. The 175-hp version of the SFT-12 engine was able to accomplish the pass in only 9.7 sec and 980 ft.

Figure 49 illustrates the results for the DOT 5%-grade speed requirements. This OAP design goal specified that the minimum sustained speed on a 5% grade with vehicle weight equal to W_{gross} shall be 70 mph for a period of 100 sec, when preceded and followed by continuous operation at 60 mph on the same grade. The performance results for this criterion are presented in terms of the speed attained after 30 sec of full-power acceleration from 70 mph on the 5% grade. Vehicle weight variations for this case were taken to be ± 15% of W_{test}; that is,

$$\text{Vehicle Weight (lb)} = W_{test} \times (1.0 \pm 0.15) + 700 \text{ lb}$$

The results show that all vehicles in all variations are able to meet the design goal with a surplus of power. Under the simulated conditions (maximum power from 70 mph on the grade), the vehicles asymptotically approach a terminal speed, about 95% of which is reached in 30 sec. This means that the maximum speed for each vehicle on the 5% grade is about 2 to 5 mph greater than the speeds indicated in Fig. 49.

The results for the 30% grade speed requirement are shown in Fig. 50. The OAP goal requires attaining and sustaining a 15-mph speed on the 30% grade. Results are presented in terms of the speed attained after 20 sec of full-power acceleration on the grade. All vehicles and all variations easily meet the 15 mph requirement and are able to accelerate from rest to that speed.

The ability of the vehicles to sustain the OAP 5% and 30% grade speeds could not be determined within the scope of the computer analysis, but sufficient capability to accomplish these requirements is included in the engine designs.

10 SEC ACCELERATION DISTANCE (FROM STANDSTILL)

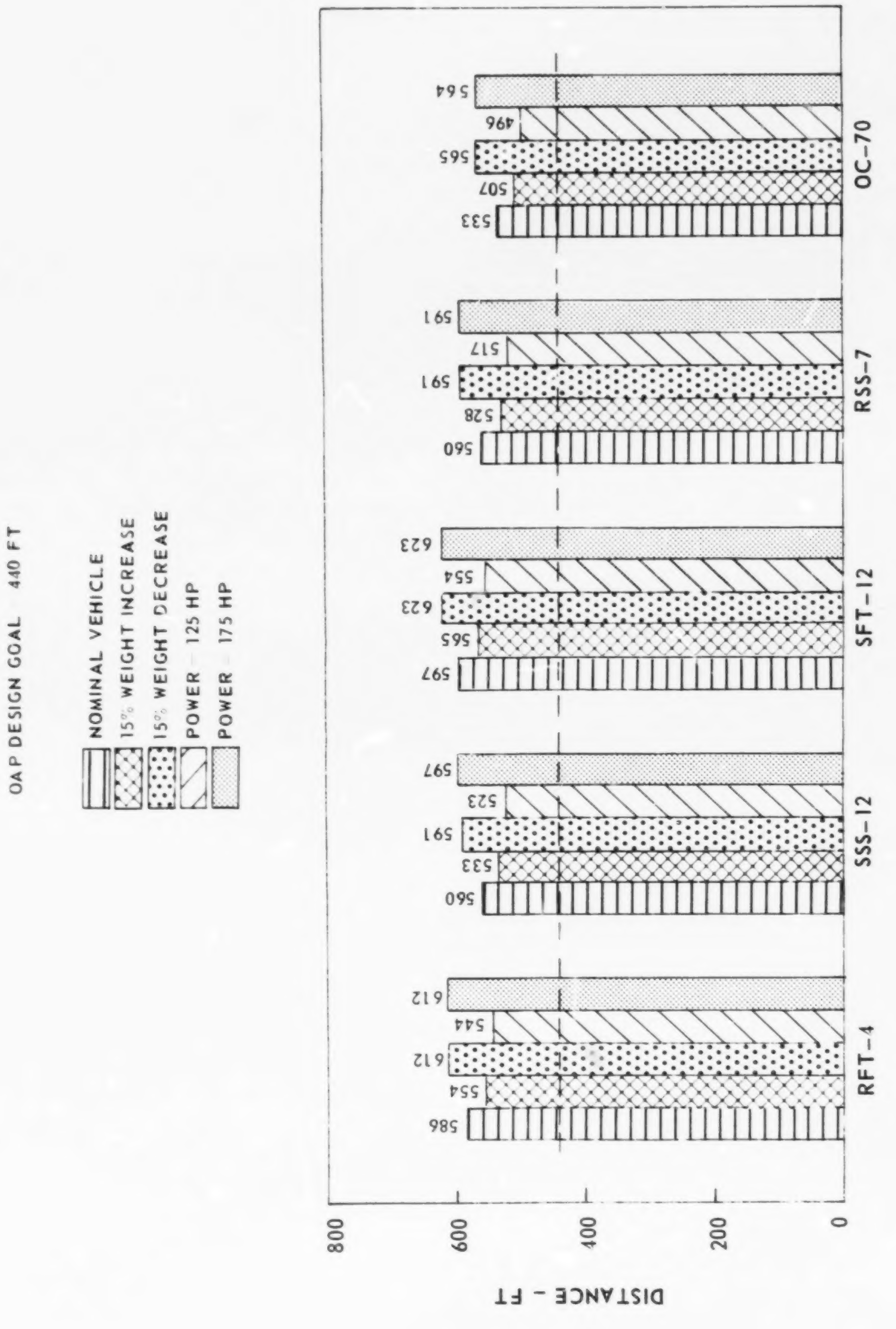


FIG. 45

0-60 MPH ACCELERATION TIME

OAP DESIGN GOAL = 13.5 SEC

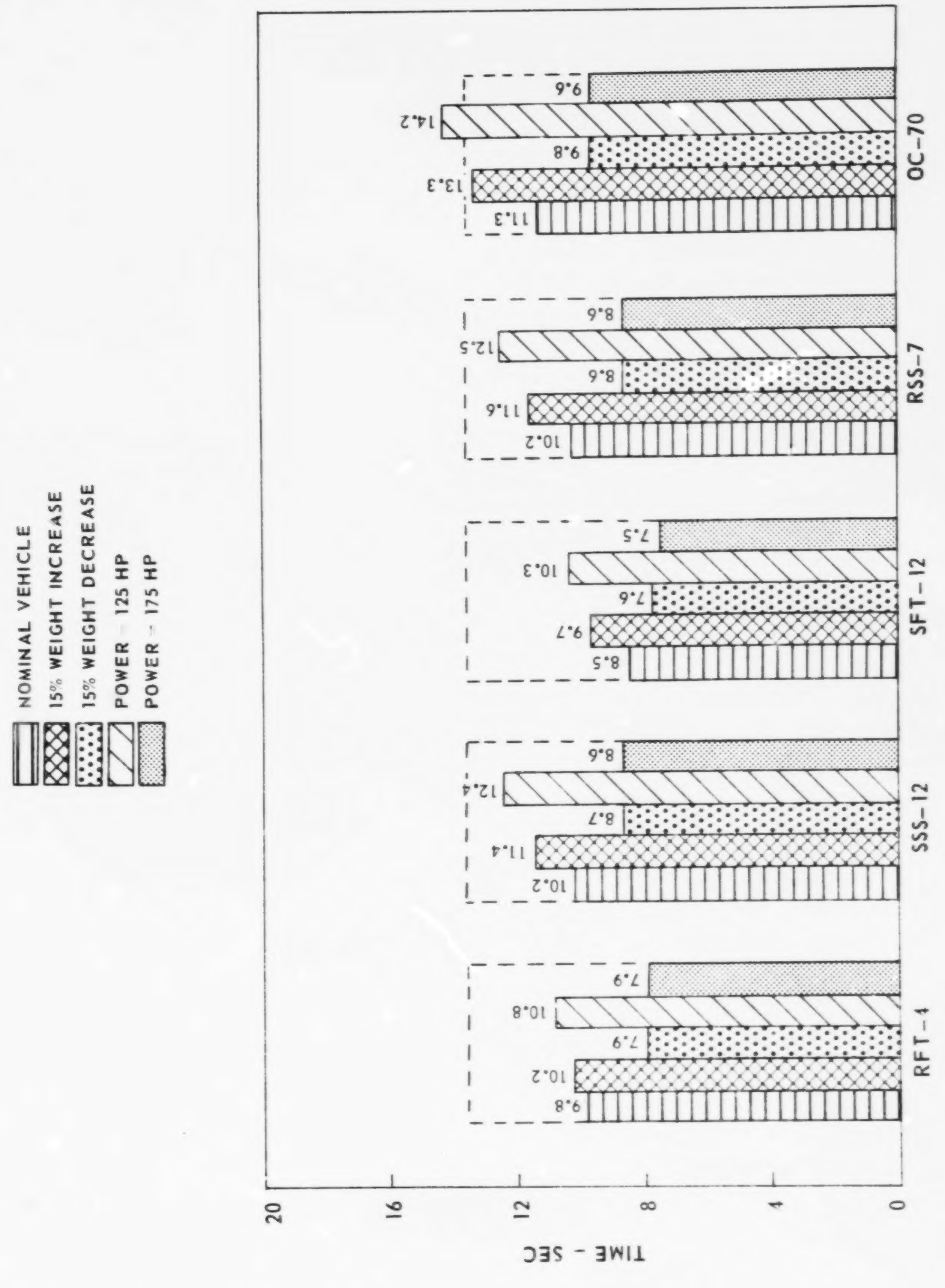
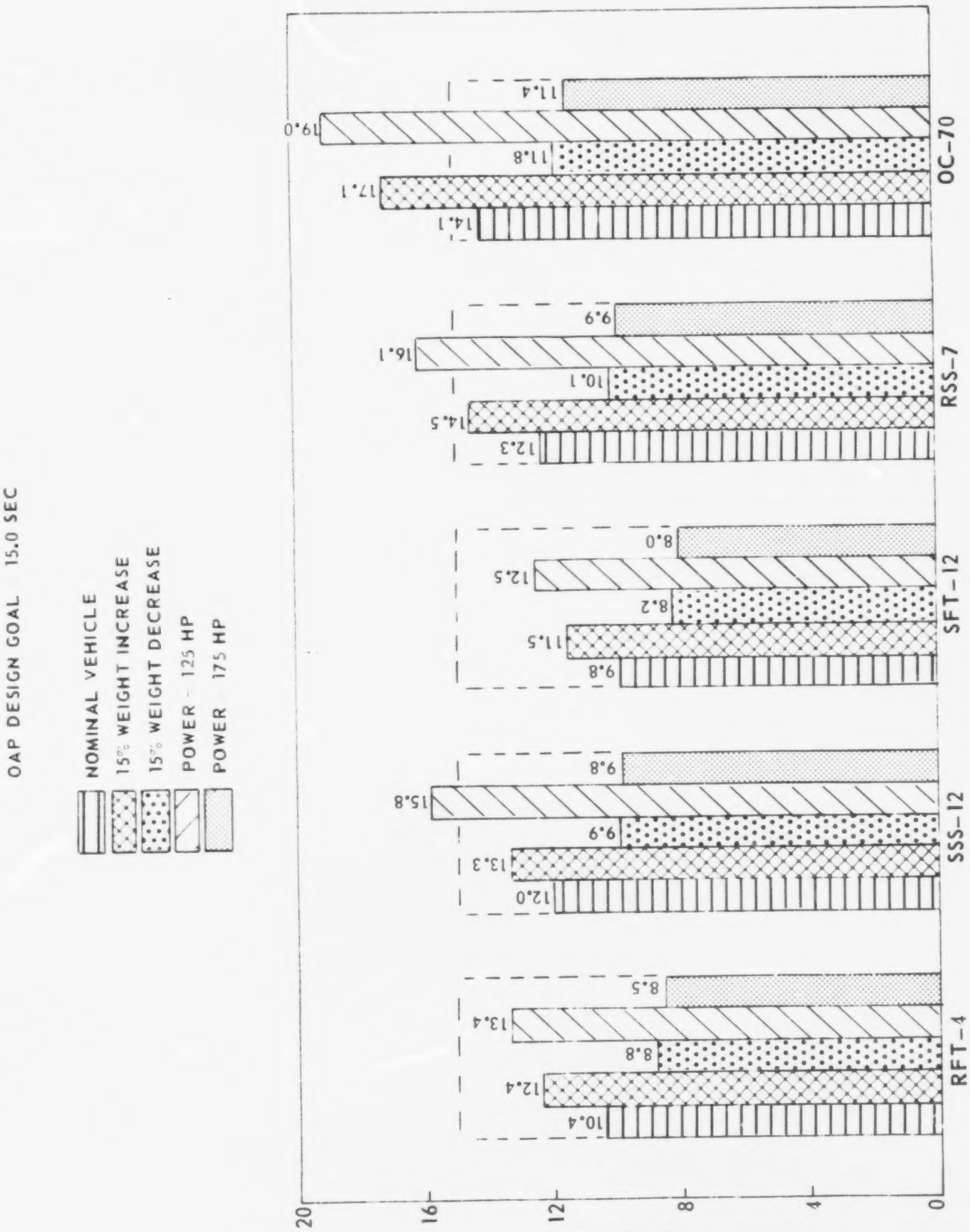


FIG. 46

25-70 MPH ACCELERATION TIME



DOT HIGH SPEED PASS MANEUVER – MAXIMUM PASSING TIME

OAP DESIGN GOAL = 15.0 SEC

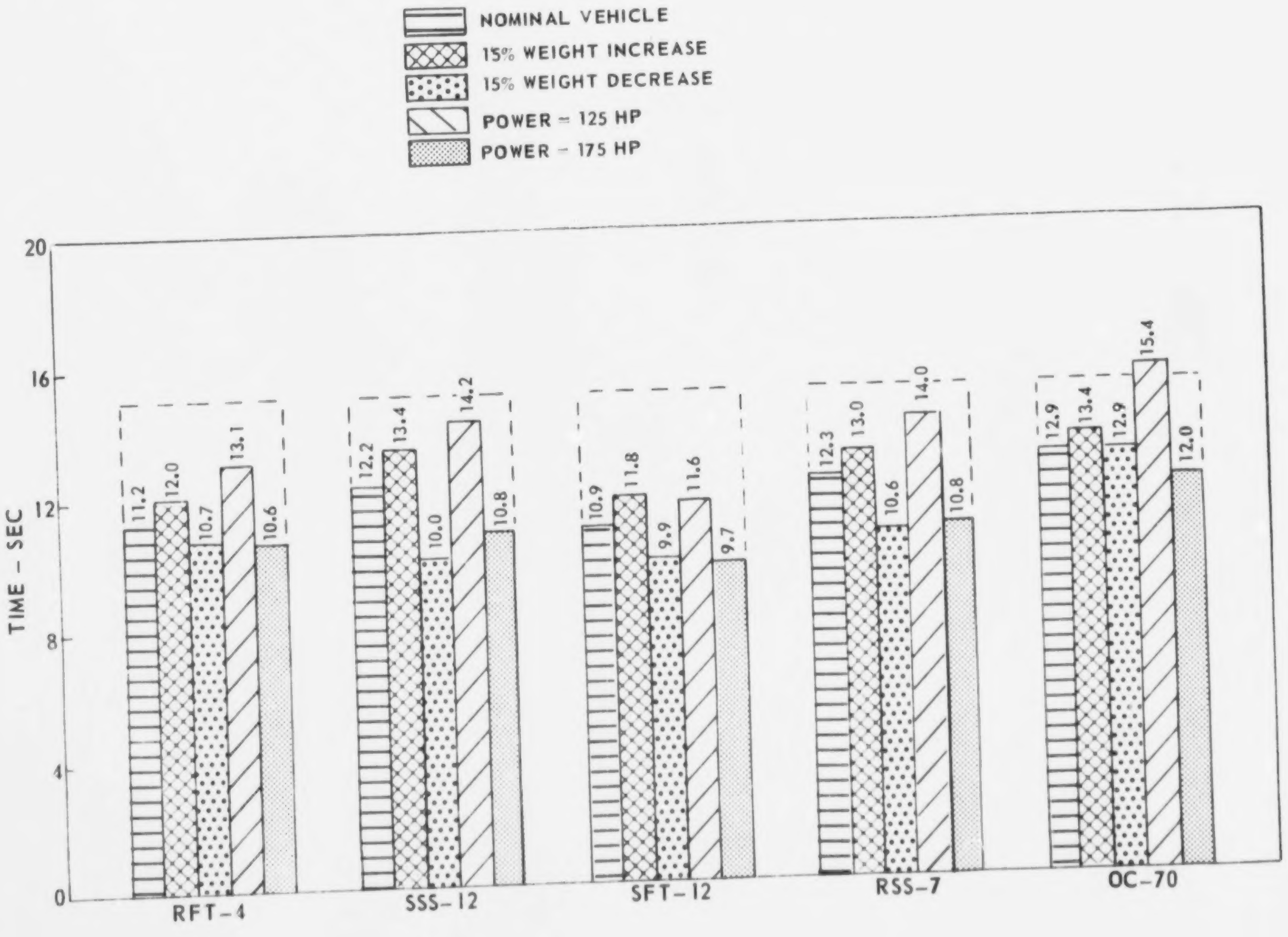
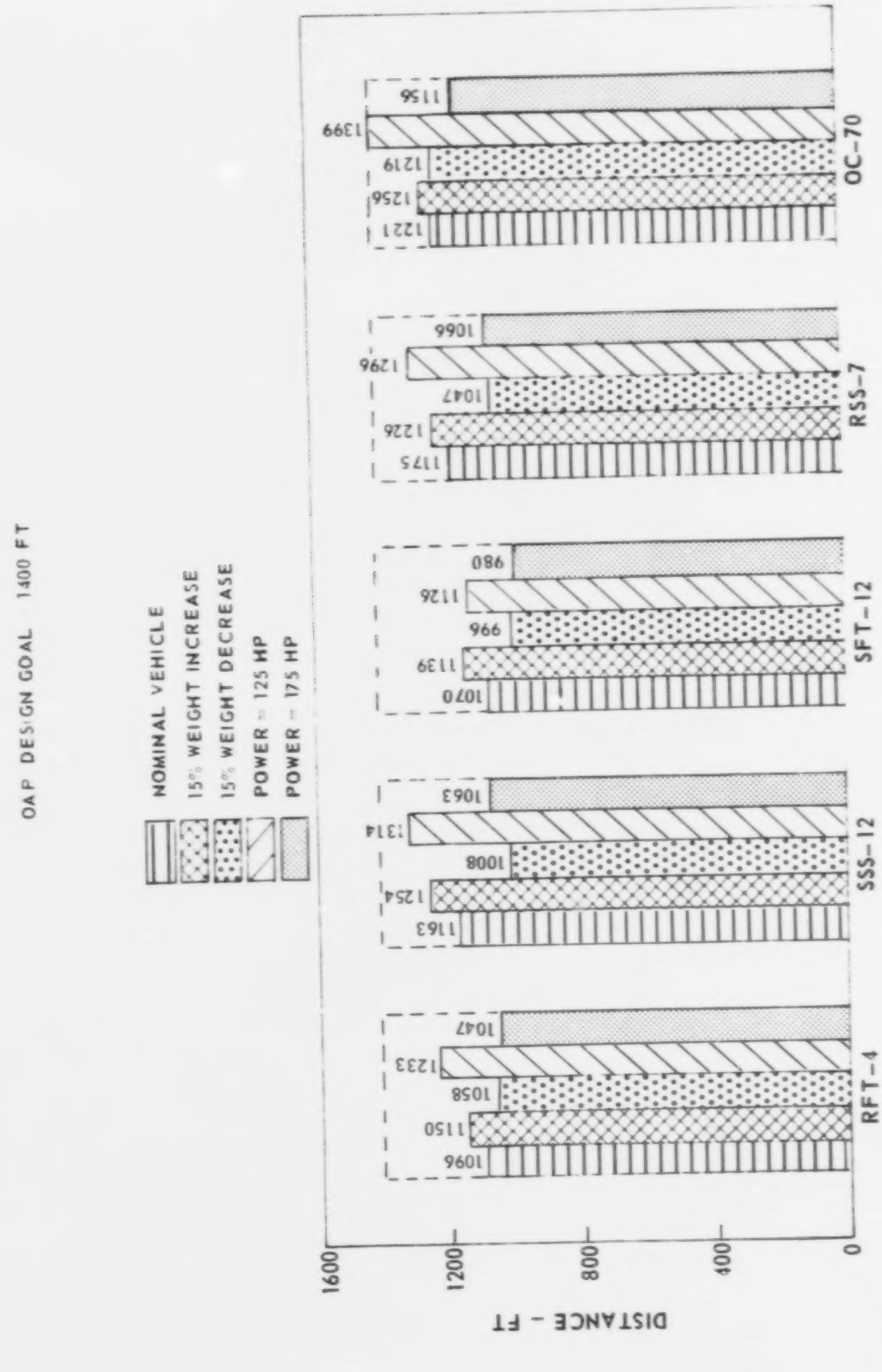


FIG. 47

DOT HIGH SPEED PASS MANEUVER – MAXIMUM PASSING DISTANCE



DOT 5% GRADE SPEED REQUIREMENTS

SPEED ATTAINED AFTER 30 SEC OF FULL POWER

ACCELERATION FROM 70 MPH ON 5% GRADE

OAP DESIGN GOAL = 70 MPH

MINIMUM SPEED ON 5% GRADE

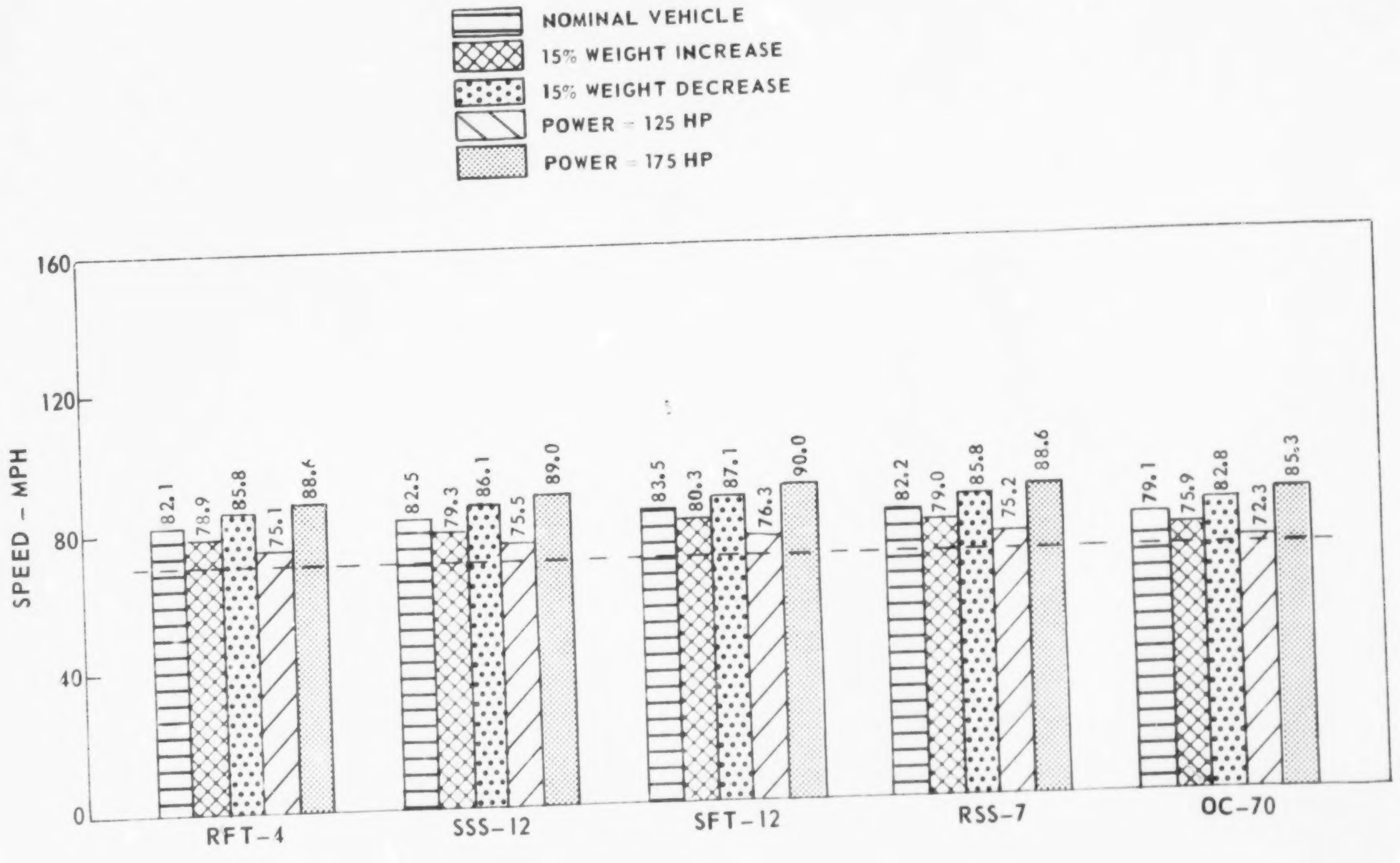


FIG. 49

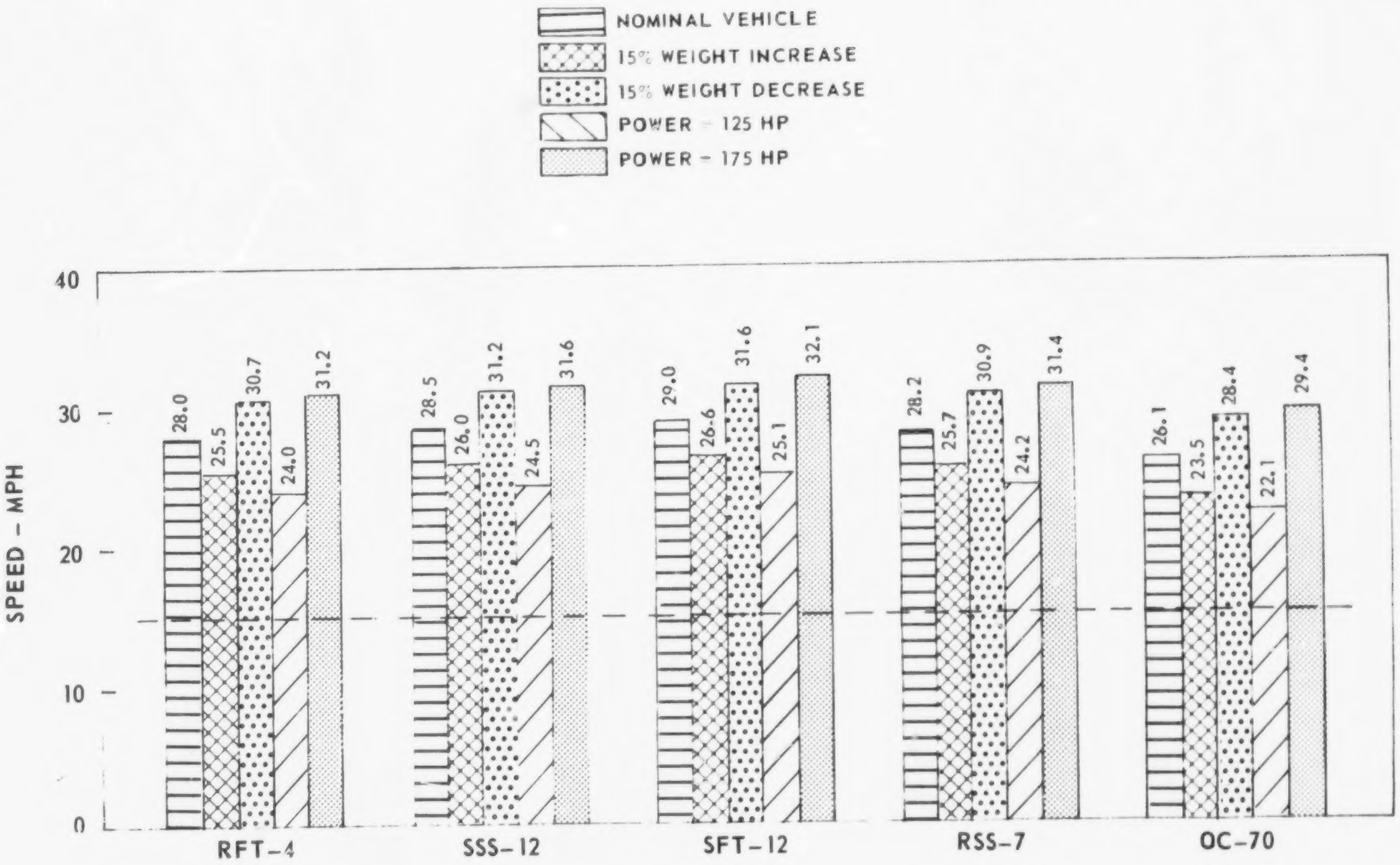
DOT 30% GRADE SPEED REQUIREMENTS

SPEED ATTAINED AFTER 10 SEC OF FULL POWER

ACCELERATION FROM STANDSTILL ON 30% GRADE

OAP DESIGN GOAL 15 MPH

MINIMUM SPEED ON 30% GRADE



It is of interest to use the results of the analysis of the effect of installed power variations in the attainment of OAP performance goals (Figs. 44 through 50) to determine the minimum installed power required to meet these goals. In the case of the single-shaft engines, the most critical acceleration performance criterion is that of the 25-to-70 mph acceleration time (Fig. 46), since this is the only requirement which cannot be met with a 125-hp engine. Interpolation of the results of Fig. 46 indicate that a minimum of 130 hp would be required for the SSS-12 or RSS-7 powered automobiles to meet the 15.0-second goal. Although this requirement is also the most critical for the free-turbine engines, an installed power of 115 hp for the RFT-4 engine and 110 hp for the SFT-12 engine are indicated to meet the minimum requirement. It must be noted, however, that extrapolation of results beyond the 125 to 175 hp range considered in this study may not be justified, since performances and weights may not be scalable beyond these limits.

Summary of Performance Analysis

The computer analysis has shown that, in general, gas turbine-powered vehicles possess an inherent performance advantage over Otto-cycle engine vehicles because of the lighter weight of the gas turbine engine. The fuel economy of the gas turbine engines, under typical driving conditions, has generally been shown to be approximately equal to, or even better than, that of Otto-cycle engines. Furthermore, any fuel economy advantage that the 1970 Otto-cycle engine may now have over the gas turbine can be expected to dwindle in years to come as the various pollution control techniques (such as reduced compression ratio, exhaust gas recirculation, and catalytic converters) decrease overall Otto-cycle engine efficiency. On the other hand, future developments in small gas turbine engine technology can be expected to improve gas turbine fuel economy as the state of the art advances.

The performance of the base-line SSS-12 engine (150 hp, 160 lb) with the 8-speed transmission relative to the current OAP goals is summarized in Table IX.

TABLE IX

PERFORMANCE RELATIVE TO OAP GOALS

	SSS-12	OAP Goal
A. Startup - time to 65% power	15-30 sec	45 sec (max.)
B. Idle operation		
Fuel consumption rate	16% of max.	14% of max.
Transmission output torque	0	40 ft lb (max.)
C. Acceleration		
10.0 sec	560 ft	440 ft (max.)
0-60 mph	10.2 sec	13.5 sec (max.)
D. Acceleration in merging traffic 25 to 70 mph	12.0 sec	15.0 sec (max.)
E. Acceleration, DOT high-speed pass maneuver	12.2 sec 1163 ft	15.0 sec (max.) 1400 ft (max.)
F. Grade velocity - min. cruise (5% grade)	83 mph	60-70 mph (min.)
G. Top Speed (zero grade)	105 mph	85 mph (min.)

ANALYSIS OF DIRECT MANUFACTURING COSTS

Definitions

Direct manufacturing costs are defined, for the purposes of this report, as:

(1) the costs of the materials used,

(2) the cost of finished or semi-finished parts bought, and

(3) the labor hours required specifically to manufacture parts and to assemble the engine, multiplied by the direct labor hourly pay rate.

Many other costs associated with the manufacturing of an engine are not accounted for in the above definition of direct manufacturing cost. These other costs include employee fringe benefits, supervision, training, production control, quality control, plant maintenance, shop supplies, light, heat, power, water, development and amortization of production tooling, rent, plant depreciation, research, development, production/industrial engineering, general administration, warehousing, distribution, product support, advertising, sales, public relations, interest, and taxes. All of these indirect costs have to be allocated (apportioned) to the products sold in addition to the direct manufacturing cost in order to establish the manufacturer's cost of doing business.

In general these costs may be allocated by burdening the direct manufacturing cost (or its elements) appropriately. These burdens depend on the specific accounting system used (i.e., how the various indirect costs are assigned to various products via appropriate overhead or burden rates) and on the specific manner in which a given manufacturer operates his business (such as make-or-buy policies, product mix, etc.). Consequently, although they are similar within a given industry, they vary somewhat among manufacturers. Furthermore, they are considered highly proprietary (for competitive reasons), and it is therefore difficult to estimate them accurately. Since an individual manufacturer clearly knows these factors as they apply to his operation, and can therefore calculate his total costs from the three direct-cost elements used herein, presentation of the results of the manufacturing cost analysis in terms of direct manufacturing cost was felt to offer the greatest utility and would involve the fewest assumptions. For rough comparisons of cost to the consumer, a lumped burden will be used in a later section.

Methodology

The automotive industry characteristically employs machines particularly suited for high-volume, specialty operations. In this study, the use of identical or similar machines and production techniques was assumed in the production of all of the automobile gas turbine engines considered herein. Under these conditions, production costs of practical hardware tend to approach the material costs of the engine itself, with direct labor costs accounting for only a small fraction of manufacturing costs. Job-shop techniques, such as those which are commonly associated with the production of aircraft gas turbines, are not only incapable of handling the production volumes necessary, but also would result in higher costs than those achievable with automatic machining methods.

The basic costing approach (following Ref. 23) was to estimate the manufacturing costs of major items individually from the detail or assembly drawings, and the minor items (standard nuts, bolts, bearings, etc.) either individually or as a small, fixed fraction of the total cost. A description of those parts which were estimated in detail, and a discussion of some of the specific considerations and assumptions used in the cost analysis, is given in the following paragraphs.

Except where indicated otherwise, drawings and specifications of major component items were furnished to experienced specialty vendors to the automotive industry (and/or aircraft-engine industry, where appropriate) for cost estimates at the two production levels of interest, i.e., 100,000 and 1,000,000 units annually. These drawings are included in Appendix IV of this report. The ground rule was to consider the most economical appropriate production method for each item, to amortize any new tooling, process development, etc., over an appropriate time period at the stipulated production rates, and to include this amortization in the estimates. The vendors were also asked to suggest design or specification changes which would reduce the manufacturing costs significantly. Where possible, several independent estimates were obtained, and any significant differences reconciled. Suggested changes were reviewed, and where found acceptable, new cost estimates were obtained.

For the simple-cycle, single-shaft engine, UACL prepared an independent labor-cost analysis based on detailed process sheets for each major part, and for automotive-type production processes, partly to check the other estimates, and partly to obtain a better definition of the labor requirements*.

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* This analysis was performed in parallel with the study, although not specifically required by the terms of the contract.

After these analyses were completed, an estimate was made of the critical materials requirement (discussed in a later section), which indicated that the originally specified materials would place a severe strain on the existing resources of some of these materials if gas turbines were to replace all spark-ignition engines. Accordingly, substitute (nickel-based) materials were identified, and the associated cost change was estimated.

It was assumed that, at the 1,000,000-unit annual rate, high-ductility forging techniques would be used to produce the impeller and turbine wheels to nearly finished dimensions from the relatively expensive materials specified. Several proprietary versions of the basic techniques have been and are being developed by United Aircraft (for current military aircraft engine programs), by certain vendors supplying United Aircraft, and elsewhere. The United Aircraft process, developed at the Florida Research and Development Center of Pratt & Whitney Aircraft, is referred to as the GATORIZING$_{TM}$ forging process (described in Refs. 24, 25, and 26). It utilizes appropriate combinations of pressure and temperature to create a high-ductility condition in powdered-metal preforms, thereby permitting them to be forged in precision dies into thin, highly convoluted parts with near-finished dimensions. The forces (and consequently press capacities) used are much lower than those normally required. This process eliminates most of the machining and material waste associated with current forged-and-machined production processes. It is expected that high-ductility forgings will be commercially available by 1975 or 1976.

For the present costing purposes, it was assumed that the cost of finished high-ductility forged parts would be \$7/lb to \$8/lb, based on the assumptions that: (1) die life and costs would be approximately the same as those presently experienced with ferrous die casting, (2) powdered-metal preforms would be used, (3) reject and scrap rate would be about 10%, (4) forming and (machine) finishing would require about 3 minutes per pound, and (5) the required tooling capital investments would be amortized in the same way as all other production tooling (i.e., via burdens applied to direct cost elements) without changing the applicable burdens significantly.

Preliminary make-or-buy decisions were made to define the basis on which the various parts should be costed, lowest cost being the sole criterion. The final decisions would have to be made on the basis of final drawings by the organization charged with the responsibility for producing the engine; it, alone, is qualified to judge the many factors which go into the individual decisions. However, since cost is generally the overriding consideration, the cost figures presented here should be representative of the final values.

All estimates were made in 1970 dollars, thus ignoring any inflation between 1970 and 1976, the initial production date considered in the study.

The basic output concerning the labor costs associated with manufacturing these engines is considered to be the number of hours required. As listed hereinafter, they represent floor-to-floor labor times* and include allowance for fatigue, handling, setup, waiting, operator efficiency, and training. They should permit any manufacturer to apply his own average labor rates to perform a first-order estimate of direct and total costs. For illustrative purposes, however, a representative value of $4/hr was assumed as a direct labor rate to permit the combination of labor with the other direct cost elements, which are expressed in dollars. It should be noted that, in view of the low labor content in the production of these gas turbines, their total direct manufacturing costs can be expected to be less sensitive to the labor rate assumed than are those for reciprocating engines.

Results of Direct Manufacturing Cost Analysis

The results of the direct manufacturing cost analysis conducted as part of this study are presented in Tables X through XVII on pages 120 through 127. The first four tables (X-XIII) present a breakdown by major component of the four basic engines (less accessories) in terms of material (or purchased-part) cost and labor minutes. The second four tables (XIV-XVII) present a direct manufacturing cost estimate for the complete engine (including accessories), where labor minutes are converted to direct cost at the rate of $4 per hour. The second set of tables includes the basic engine data reflected in the first set of tables, and also includes r-m-s (root-mean-square) uncertainty estimates. The uncertainty-estimate derivation is discussed on pages 128 to 131.

SSS-12

Each part was costed individually for the simple-cycle, single-shaft engine. Estimates were made of the labor minutes required to manufacture parts from raw material, to finish parts furnished by suppliers, and to assemble the engine. Material and purchased-part costs were estimated as described earlier. Accessories (starter, alternator, voltage regulator, and oil pump), were assumed to be similar to the units now used in automobiles, and were estimated at OEM (original equipment manufacturer) purchase prices. Standard items (nuts, bolts, O-ring seals, shims, etc.) were also estimated at OEM purchase prices. The cost of the fuel control (including fuel pump) was based on the assumption that the fuel control would be hydromechanical in nature, and was extrapolated from information supplied by a major supplier of automotive carburetors and gas turbine controls.

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* The term "floor-to-floor labor time" covers that period of time during which the operator takes a part from the floor (table, bench, etc.), places it in the machine, performs the required operation(s), and returns it to the floor, or to its next station.

The basic costs are shown for each part costing more than $2 in Table X; lower-cost parts are lumped together under "Miscellaneous." Costs are totaled in terms of labor minutes and material dollars, which include both materials and purchased parts; purchased parts are identified by asterisks in the table. These totals are also given in Table XIV, with the material and purchased-part costs now identified separately. The accessory costs are listed with the fuel control system (including fuel pump) identified separately. All costs are summed in a total direct cost (with direct labor at $4/hr). The major direct cost elements for the SSS-12 (rounded off to hours and dollars) are:

Production rate, units annually		100,000		1,000,000
Labor @ $4/hr		$ 12		$ 7
Materials		165		108
Purchased parts		210		161
Basic Engine	$ 61		$43	
Fuel Control System	106		75	
Other Accessories	43		43	
Total Direct Manufacturing Cost		$387		$276

Estimated uncertainty limits for these costs are also shown in Table XIV; they are discussed in a later section.

SFT-12

For the simple-cycle, free-turbine engine, every part was also costed individually (using estimates previously made for the single-shaft engine in the case of identical parts, and scaling previously made estimates according to weight, in the case of similar parts). The estimates for fuel control system and accessories were the same as those for the single-shaft engine, since it was assumed that the fixed geometry in the power turbine section of the free-turbine engine would not increase the control requirements for this engine to any significant degree, although additional control complexity may be required in order to provide competitive acceleration and acceptable braking characteristics. The results are shown in Tables XI and XV.

RSS-7

Costs for the regenerated single-shaft 7:1-pressure-ratio engine were not estimated in the same detail as the simple-cycle engines. Only parts judged to cost over $2 were costed individually. The required dimensions were obtained from

TABLE X

LABOR AND MATERIAL FOR MAJOR COMPONENTS OF BASIC ENGINE

SSS-12

Part No. (see Fig. 1)	Part or Operation	Material	Finished Weight, lb	Annual Production Rate 10^5 Labor, min	Annual Production Rate 10^5 Material, $	Annual Production Rate 10^6 Labor, min	Annual Production Rate 10^6 Material, $
1	Assembly of Star, Exducer, and Shaft	--	--	14.5	0	8.5	0
2	Turbine Rotor Star	Udimet 700	0.95	12.5	29.90	3.0	6.80
3	Turbine Rotor Exducer	Coated IN 100	0.5	1.0	11.50	1.5	3.40
4	Turbine Shaft	SAE 4340 (tube)	0.33	5.0	1.00	4.3	0.90
5*	Turbine Shroud (Rear)	IN 738	1.71	0	10.00	0	8.55
9	Combustor Scroll	Hastelloy-X	1.42	5.2	10.10	4.0	9.00
11	Turbine Nozzle	Coated WI 52 (Alternative: IN 738)	3.5	2.5	30.00	1.5	20.00
15	Combustion Can	Hastelloy-X	0.35	1.7	10.00	0	8.55
19*	Turbine Shroud (Front)	Coated WI 52 (Alternative: IN 738)	0.75	0	10.00	0	6.00
21, 22	Assembly of Shaft, Inducer, and Impeller	--	--	27.5	0	11.3	0
23	Compressor Shaft	SAE 4340	0.45	4.9	1.00	2.7	0.90
24	Compressor Inducer	Custom 455	0.49	4.3	6.00	1.5	3.35
25	Compressor Impeller	AMS 4928	0.82	29.0	17.00	2.5	5.65
26, 27, 28	Diffuser Case	Cast Iron	19.7	17.9	9.60	16.0	8.65
40, 53	Angular Contact Bearings (2)	52CB	0.3	0	12.00	0	12.00
44	Reduction Gear Box Case	Cast Iron	29.6	3.2	17.80	3.2	16.05
45	Reduction Gear	Steel	5.6	19.2	4.50	15.2	4.05
52, 54*	Shaft Seals (1)	Steel/Carbon	0.3	0	4.00	0	2.00
	(2)		0.3	0	3.00	0	1.50
**	Miscellaneous	--	--	11.6	25.70	9.1	21.60
	Final Assembly and Test	--	--	23.8	0	21.4	0
	TOTAL: BASIC ENGINE SSS-12			187	226	110	151

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* Purchased part

** Including purchased parts

TABLE XI

LABOR AND MATERIAL FOR MAJOR COMPONENTS OF BASIC ENGINE

SPT-12

Part No. (see Fig. 10)	Part or Operation	Material	Finished Weight, lb	Annual Production Rate 10⁵ Labor, min	Annual Production Rate 10⁵ Material, $	Annual Production Rate 10⁶ Labor, min	Annual Production Rate 10⁶ Material, $
1	Turbine Rotor - Gas Generator	Udimet 700	1.5	14.1	33.00	4.5	10.20
2	Assembly of Rotor and Shaft	--	--	14.5	0	8.5	0
3	Turbine Shaft	SAE 4340	0.33	5.0	1.00	4.3	0.90
3	Compressor Shaft	SAE 4340	0.45	4.9	1.00	2.7	0.90
4*	Turbine Shroud (Rear)	IN 738	1.7	0	10.00	0	8.55
8	Combustor Scroll	Hastelloy-X	1.4	5.2	10.10	4.0	9.00
9	Turbine Nozzle - Gas Generator	Coated WI 52 (Alternative: IN 738)	3.3	2.5	29.00	1.5	19.00
17*	Turbine Shroud (Front)	Coated WI 52 (Alternative: IN 738)	0.75	0	10.00	0	6.00
19	Assembly of Shaft, Inducer, and Impeller	--	--	27.5	0	11.3	0
20	Compressor Impeller	AMS 4928	0.8	29.0	17.00	2.5	5.65
21	Compressor Inducer	Custom 455	0.5	4.3	6.00	1.5	3.35
22, 23, 24	Diffuser Case	Cast Iron	19.7	17.9	9.60	16.0	8.65
36	Accessory Drive Gear Case	Cast Iron	13.6	4.0	6.10	4.0	5.50
37	Pinion Gear	Steel	5.6	19.2	4.30	16.2	4.05
41	Air Inlet Case	Cast Iron	27.0	4.3	16.20	3.3	14.60
49	Compressor Shroud	Cast Iron	5.5	4.3	2.60	3.5	2.20
52, 53, 68*	Shaft Seals (1)	Steel/Carbon	0.3	0	3.00	0	1.50
	(2)	Steel/Carbon	0.3	0	3.00	0	1.50
	(3)	Steel/Carbon	0.3	0	4.00	0	2.00
55	Combustion Casing	Cast Iron	18.5	5.8	11.20	2.0	10.10
56	Housing - Power Turbine Rotor	Cast Iron	20.0	4.1	13.20	3.6	9.00
57	Exhaust Case	Cast Iron	10.1	5.0	8.40	4.4	8.05
58, 59	Nozzle - Power Turbine	IN 706 (Alternative: IN 100)	4.0	4.0	35.00	3.5	23.40
61, 62*	Rotor - Power Turbine	IN 100	1.7	5.0	26.00	4.0	16.75
63*	Gear Shaft - Power Turbine	Steel	0.75	0	16.40	0	13.80
64, 67*	Angular Contact Bearings (four required)	52CB	0.25	0	24.00	0	24.00
71	Reduction Gear - Box Case	Cast Iron	15.0	3.2	8.10	3.2	7.30
77	Output Gear and Shaft	Steel Plate and SAE 4340 (tube)	4.4	17.7	4.15	17.0	3.75
78*	Tapered Roller Bearings (two required)	52CB	0.6	0	6.00	0	6.00
**	Miscellaneous	--	--	19.7	39.60	19.0	33.75
	Final Assembly and Test	--	--	35.0	0	31.0	0
	TOTAL: BASIC ENGINE SPT-12			250	358	172	260

---

* Purchased part

** Including purchased parts

TABLE XII

LABOR AND MATERIAL FOR MAJOR COMPONENTS OF BASIC ENGINE

RSS-7

Part No. (see Fig. 13)	Part or Operation	Material	Finished Weight, lb	Annual Production Rate 10^5 Labor, min	10^5 Material, $	10^6 Labor, min	10^6 Material, $
1	Inlet Air Duct	Cast Iron	13.6	4.5	8.20	3.7	7.35
2	Front Cover for Accessory Drive Case	Cast Iron	16.4	4.5	9.80	4.2	8.50
3	Compressor Shroud	Cast Iron	9.2	3.5	3.70	2.5	3.35
4	Compressor Impeller	Steel	1.9	23.0	6.30	5.9	8.40
5	Main Drive Shaft and Pinion Gear	Gear Steel and Tube	0.5	7.0	1.00	4.3	0.90
6	Gear and Bearing Yoke	Steel	9.8	8.5	18.40	6.5	16.55
7	Idler Gear (Off Main Shaft)	Gear Steel	1.6	9.0	1.00	7.5	0.90
8	Internal Gear Ring (On Yoke)	Gear Steel	2.0	3.2	3.00	1.4	2.70
9	Starter Drive Gear	Gear Steel	0.4	3.1	0.80	1.4	0.70
10	Accessory Drive Gear	Gear Steel	0.35	2.8	0.70	2.5	0.65
11	Accessory Pinion Gear and Drive Gear (Including Shaft)	Gear Steel and Tube	5.4	11.1	7.80	7.8	7.65
12	Fuel Control Drive Gear and Shaft	Gear Steel and Rod	1.6	9.0	1.00	7.5	0.90
13	Diffuser Case	Cast Iron	24.8	22.0	12.40	19.2	11.90
14	Burner Liner	Hastelloy-X	0.6	2.3	10.00	2.3	8.00
15	Combustor Scroll	HS-188	2.2	6.0	15.60	4.6	14.00
16*	Fuel Nozzle	--	0.4	0	5.00	0	3.50
17	Turbine Nozzle	Coated WI 52 (Alternative: IN 738)	5.2	2.8	32.40	1.8	21.60
18	Turbine Rotor	Udimet 700	1.8	17.5	70.50	6.3	11.90
19*	Turbine Shroud	IN 738	0.9	0	11.40	0	7.60
20	Exhaust Diffuser	Coated Cast Iron	18.0	3.0	14.60	2.7	13.20
21*	Anti-Friction Bearings (5 large; 1 small)	52CB	1.7	0	17.00	0	17.00
22	Pressure Vessel	Cast Iron	49.0	15.0	29.05	13.0	23.20
23**	Regenerator (Including Carrier, Seals, and Screen Pack)	--	65.0	43.5	116.75	24.0	105.70
24*	Regenerator Drive Motor	--	10.0	0	13.00	0	11.00
**	Miscellaneous	--	--	3.0	36.40	8.5	15.15
	Final Assembly and Test	--	--	38.0	0	33.0	0
	TOTAL: BASIC ENGINE RSS-7			250	446	171	323

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* Purchased part

** Includes purchased parts

TABLE XIII

LABOR AND MATERIAL FOR MAJOR COMPONENTS OF BASIC ENGINE

RPE-4

Part No. (see Fig. 16)	Part or Operation	Material	Finished Weight, lb	Annual Production Rate 10^5 Labor, min	Annual Production Rate 10^5 Material, $	Annual Production Rate 10^6 Labor, min	Annual Production Rate 10^6 Material, $
1	Air Inlet Housing	Cast Iron	14.0	3.5	8.40	3.2	7.55
2	Accessory Drive Front Cover	Cast Iron	15.8	3.5	9.60	3.3	6.40
3	Compressor Shroud	Cast Iron	6.1	4.0	4.40	3.5	4.00
4	Compressor Impeller	Aluminum	1.7	7.0	10.50	6.9	8.65
5	Compressor Drive Shaft and Gear	Steel	1.0	5.0	0.70	2.5	0.65
6	Gear for Fuel Control Drive	Gear Steel	3.1	11.0	2.20	10.0	1.55
7	Starter Drive Gear	Gear Steel and Tube	1.6	9.0	1.00	7.5	0.90
8	Accessory Drive Gear	Gear Steel and Rod	1.9	14.6	4.60	8.6	4.10
9	Diffuser Case	Cast Iron	35.2	20.0	17.60	19.2	16.85
10*	Regenerator Disk	Cercor	20.0	0	50.00	0	37.00
11	Burner Liner	Hastelloy-X	0.8	2.3	10.00	2.3	9.00
12	Combustor Scroll	Hastelloy-X	2.2	6.0	15.60	4.6	14.00
13*	Fuel Nozzle	--	0.4	0	5.00	0	3.50
14	Turbine Nozzle - Gas Generator	Coated WI 52 (Alternative: IN 738)	5.2	5.0	37.50	1.9	25.00
15	Turbine Rotor - Gas Generator	Udimet 700	3.1	17.8	70.50	5.8	15.20
16*	Turbine Shroud	Coated WI 52 (Alternative: IN 738)	3.4	0	11.40	0	7.60
17	Nozzle - Power Turbine	IN [illegible]	8.0	25.0	78.10	13.0	63.13
18*	Rotor - Power Turbine	IN 100	2.3	5.0	95.40	4.0	22.74
19	Output Shaft and Pinion	Steel/Gear Steel	[illegible]	7.0	1.00	2.5	0.90
20	Exhaust Diffuser	Cast Iron	14.8	5.3	13.30	2.0	10.95
21	Output Gears (2)	Gear Steel	[illegible]	15.8	5.60	16.1	5.00
22	Pressure Vessel	Cast Iron	38.0	[illegible]	48.30	19.5	42.85
23*	Anti-Friction Bearings (8)	52100	[illegible]	0	24.33	0	24.00
24**	Regenerator Support, Seals, Drive Gear, etc.	Cast Iron	40.0	40.0	111.85	24.0	100.50
25*	Regenerator Drive Motor	--	10.0	0	13.00	0	11.00
**	Miscellaneous	--	--	24.0	46.95	5.3	26.25
	Final Assembly and Test	--	--	21.0	0	48.5	0
	TOTAL: BASIC ENGINE RPE-4			[illegible]	645	213	469

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* Purchased part

** Includes purchased parts

TABLE XIV

DIRECT MANUFACTURING COST ESTIMATE OF COMPLETE ENGINE

SSS-12

Annual Production Rate	10^5	10^6
Labor Minutes	187	110
Basic Engine (from Table X)		
Materials	$165	$108
Purchased Parts	61	43
Subtotal: Materials and Purchased Parts	$226	$151
Direct Labor (@ $4/hr)	12	7
Subtotal: Basic Engine	$238	$158
Accessories		
Fuel Control System	$106	$ 75
Other Accessories	43	43
Total Direct Manufacturing Cost	$387	$276
Uncertainties (Dollars)		
Labor	+1 -0	+1 -0
Materials and Purchased Parts	+14 -16	+23 -11
Accessories	+25 -26	+25 -26
TOTAL	+29 -30	+34 -28

TABLE XV

DIRECT MANUFACTURING COST ESTIMATE OF COMPLETE ENGINE

SFT-12

Annual Production Rate	10^5	10^6
Labor Minutes	247	168
Basic Engine (from Table XI)		
Materials	$250	$176
Purchased Parts	108	84
Subtotal: Materials and Purchased Parts	$358	$260
Direct Labor (@ $4/hr)	16	11
Subtotal: Basic Engine	$374	$271
Accessories		
Fuel Control System	$106	$ 75
Other Accessories	43	43
Total Direct Manufacturing Cost	$523	$389
Uncertainties (Dollars)		
Labor	+1 -0	+1 -0
Materials and Purchased Parts	+21 -23	+31 -15
Accessories	+25 -26	+25 -26
TOTAL	+33 -35	+40 -30

TABLE XVI

DIRECT MANUFACTURING COST ESTIMATE OF COMPLETE ENGINE

RSS-7

Annual Production Rate	10^5	10^6
Labor Minutes	250	171
Basic Engine (from Table XII)		
Materials	$374	$270
Purchased Parts	72	53
Subtotal: Materials and Purchased Parts	$446	$323
Direct Labor (@ $4/hr)	17	11
Subtotal: Basic Engine	$463	$334
Accessories		
Fuel Control System	$106	$ 75
Other Accessories	43	43
Total Direct Manufacturing Cost	$612	$452
Uncertanties (Dollars)		
Labor	+1 -1	+1 -1
Materials and Purchased Parts	+42 -49	+64 -41
Accessories	+25 -26	+25 -26
TOTAL	+49 -55	+69 -49

TABLE XVII

DIRECT MANUFACTURING COST ESTIMATE OF COMPLETE ENGINE

RFT-4

Annual Production Rate	10^5	10^6
Labor Minutes	301	209
Basic Engine (from Table XIII)		
Materials	$470	$332
Purchased Parts	173	137
Subtotal: Materials and Purchased Parts	$643	$469
Direct Labor (@ $4/hr)	20	13
Subtotal: Basic Engine	$663	$482
Accessories		
Fuel Control System	$141	$110
Other Accessories	43	43
Total Direct Manufacturing Cost	$847	$635
Uncertainties (Dollars)		
Labor	+1	+1
	-1	-1
Materials and Purchased Parts	+44	+59
	-47	-35
Accessories	+25	+25
	-26	-26
TOTAL	+50	+64
	-54	-44

assembly (rather than detail), drawings, and weights were estimated from these dimensions. Other parts were lumped in a miscellaneous allowance amounting to about 7.5% of the total (first-iteration) engine cost at 100,000 units annually, and 5% of the total at 1,000,000 units annually (based on analysis of the simple-cycle engines and experience with similar engines; see also Ref. 23). Similarly, engine line assembly and test costs were estimated at 2.2 and 2.7%, respectively, of the total engine costs at the two production rates. (Subassembly labor was included as part of the component costs, as before.) Although the fuel control system may have to fulfill additional functions to achieve adequate engine performance, costs of the system were assumed to be the same as those for the simple-cycle engines. The resulting costs are shown in Tables XII and XVI.

RFT-4

The conventional-technology regenerated engine was costed in the same manner as the RSS-7 engine. The vanes used in the variable-geometry power turbine, which are required to obtain engine braking and relatively flat part-load performance, represent a major cost item if produced by conventional methods, as was assumed (although there is a possibility that the vanes, each with its control shaft, could be coined at about one-fifth of the conventional cost). The fuel control unit in this engine has to control the power-turbine vanes, and the cost penalty associated with this function has been estimated at about 50% and 35%, respectively, at the two production levels. The resulting costs are shown in Tables XIII and XVII.

Uncertainties

The greatest uncertainties in the above estimates stem from the fact that no gas turbine automobile engine has ever been manufactured in automobile-type production volumes, and therefore the costing background available is simply not as comprehensive as that in use for the conventional gasoline engine. This drawback is particularly severe since the parts which are quite dissimilar to present experience (rotors, regenerators, etc.) apparently account for larger proportions of the cost than do parts which are similar to present experience (such as machined iron castings). Thus, the accuracy of the cost-estimating process depends to a large degree upon speculative extrapolations of manufacturing technology which is yet to be demonstrated in high-volume production. Nevertheless, sufficient experience can be directed to the problem at hand to develop useful information on probable manufacturing costs which are sufficiently accurate to assess the overall feasibility of a gas turbine in automobiles. In other words, the current study was not intended to define the very small differences which are so important in determining actual profitability of production in the automotive industry, but rather, to define the desirability of further, more definitive studies and demonstrations.

With the above qualification in mind, confidence in the validity of the costing procedure is based, to a large measure, on the fact that the performance of all advanced-technology components has been demonstrated on the component level at conditions similar to (though not identical with) those under which they would operate in these engines. Based on these tests and experience with similar components, there is no doubt that, with some development, components very similar in geometry to those used herein will deliver substantially the performance projected for them. Consequently, engines synthesized from these components will perform substantially as predicted, and no allowance has been made in the uncertainty calculations for major design changes.

Minor changes can be considered to fall into four categories:

1. Cost-unrelated changes, such as changes in vane angles, location of bolts, etc. Most of the differences between the preliminary design and the corresponding final design(s) would probably be in this category.

2. Size changes, such as larger or smaller diameters, thicker or thinner walls, etc., leading to weight changes and, therefore, proportional cost changes. It has been assumed that all major components are subject to such changes, thereby leading to a cost uncertainty of ± 10% on each.

3. Materials and process changes

4. Accessory changes

Both materials and process changes, and accessory changes, have been considered for each major item and are discussed below.

Materials and Processes

Labor and materials uncertainties have been estimated for each part. These uncertainties involve both the likelihood of having to use more expensive materials or processes and the likelihood of being able to reduce materials and/or processing costs by an intensive production engineering effort. Generally, this approach yielded estimated uncertainties in the range of +5 to 10% and -10 to -15% of the material cost, and +10 to 15% and -0 of the labor cost of each individual item, at both production rates.

At the 1,000,000-unit annual rate, the uncertainty stemming from the applicability of high-ductility forging was estimated as -10% and +100 to 150% for each part; the upper bracket reflects the possibility that the given part cannot be forged by this process economically, so that conventional processes would have to be used. This uncertainty is, clearly, the major contributor to the materials uncertainties quoted at this production rate.

Accessories

The major uncertainty in the cost of the accessories relates to the fuel control. The specific requirements (in terms of operating parameters to be sensed, processing to be applied to the sensed information, and control action to be performed) cannot be firmly established until demonstration engines are tested, and until transmissions and engine/transmission/vehicle control interactions are more closely defined. These requirements will be further modified by customer reaction assessment, which can be obtained only through extensive road testing of complete vehicles. Furthermore, it is quite probable that several fuel control approaches, varying in cost and resulting vehicle performance, will be available as options. Such options would correspond to present-day owner options on available equipment, as best exemplified by the use of availability of four-barrel carburetors or other high-performance options. The finally selected fuel control, which satisfies the requirements at lowest total (engine plus fuel control) cost, may therefore cost more or less than the amount estimated here. The uncertainties were estimated to be ± $25 for all engines, for both production rates. An even lower limit could be possible, but this assumption would imply some optimism concerning the ability of LSI (large-scale integrated) electronic components to perform the required functions more economically, toward the end of the 1970's, than can now be done with hydromechanical devices.

As indicated previously, a possibility exists that a single device can serve both as a starter and a generator for these engines, but the relatively long start cycle of an automobile gas turbine may prove unacceptable, thus requiring a starter of greater capacity and higher cost. The overall uncertainty regarding starters was estimated at ± $7.

Combined Uncertainties

The individual uncertainties, including the ± 10% uncertainties for size or weight changes, and uncertainties in the lumped cost of minor parts (not individually estimated for the regenerated engines), were combined by rms* calculations of separate upper and lower limits (as in Ref. 27), and are shown in Tables XIV to XVII. The rms subtotals for uncertainties in labor, basic engine materials and parts, and accessories, are shown, as is the total uncertainty.

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* "rms" stands for root-mean-square, and in this case means taking the square root of sums of squares.

Combining uncertainties by rms calculations implies that the phenomena giving rise to the separate uncertainties are independent of each other. Although this assumption is clearly true in some instances, certain exceptions exist. For instance, if the impeller rotor has to be larger in diameter, the shroud has to be larger as well, and the turbine rotor and housing may also have to be larger. It may, therefore, be more appropriate to add the uncertainties in groups of parts which are likely to interact with each other, and then perform rms calculations for group totals. However, preliminary calculations indicate that this process would not increase the total uncertainty by more than $10.

It must be emphasized that the uncertainties shown here are considered valid only for the engines as designed. Redesign, optimization, and productionization procedures can change the figures quoted somewhat. Thus, the basic cost figures should be interpreted as indicative rather than definitive, and of even greater validity is the interpretation of these results as relative cost differences among the gas turbine engine designs studied, rather than as the absolute value of production costs for engines of this type.

Discussion

The results of the direct manufacturing costs estimated for complete engines are shown in Fig. 51, with shading to indicate the rms limits of calculated uncertainties. As indicated previously, for cost-estimating purposes the cost elements given individually in Tables XIV to XVII are more significant than the totals, but the figure affords a convenient visual means of comparing completely equipped engines with each other.

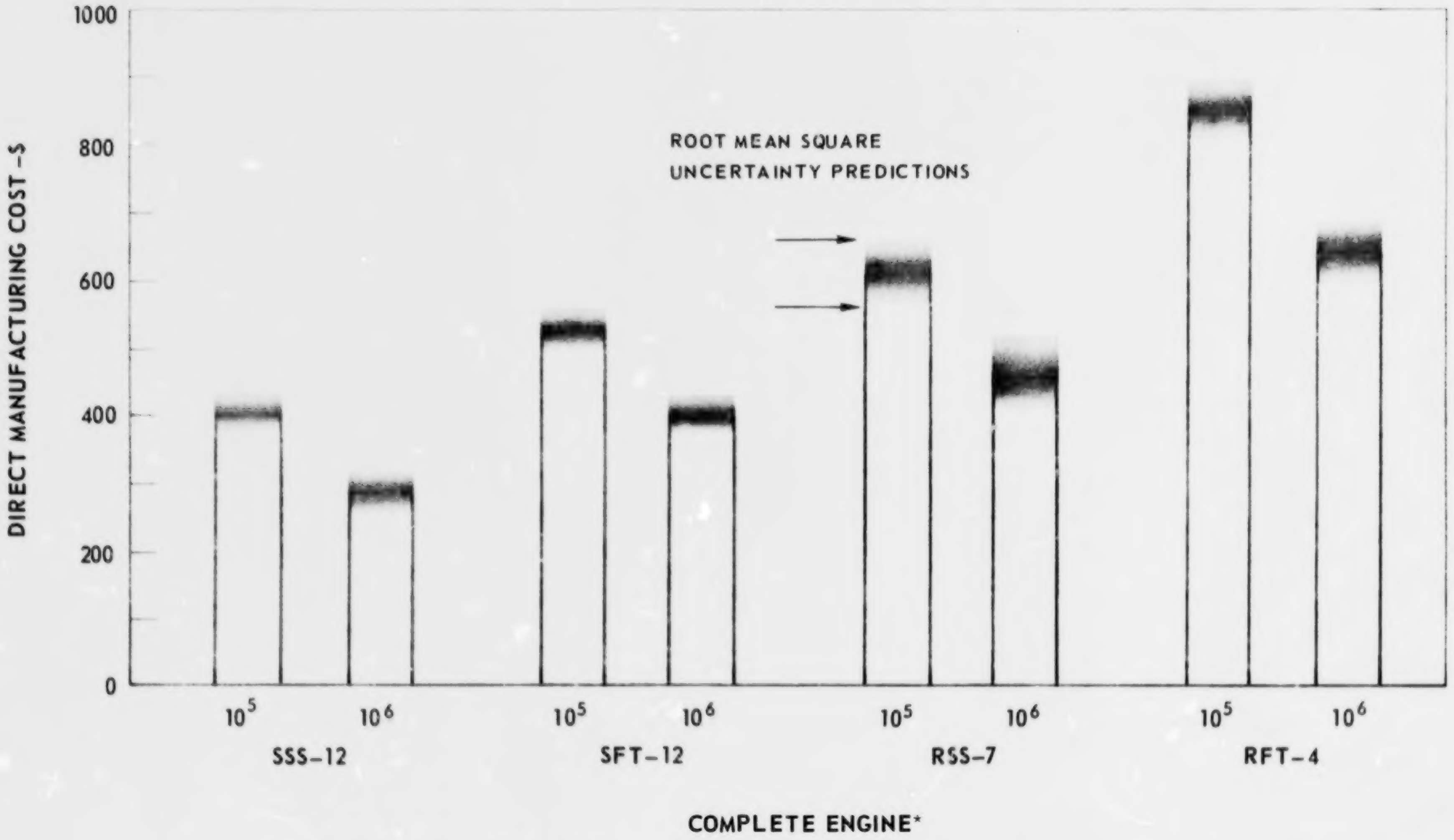
SSS-12

The costs of the simple-cycle, single-shaft engine are the lowest of any of the four types of engines studied. These costs are low because the engine is physically small, and therefore light in weight, and contains a minimum number of parts. Although the design requires high-cost, critical-material alloys in the compressor, burner, and turbine sections, these parts weigh very little. Also, as a result of the small number of parts, the total labor content (including final assembly) is low*.

---

* It should be pointed out that since several of the basic engine components (and all of the accessories) are purchases, there is additional labor contained in the basic engine over and above that costed here. This amount is estimated to be about one hour and is included in the purchased-parts costs.

SUMMARY OF DIRECT MANUFACTURING COSTS



*FUEL CONTROL (INCLUDING PUMP), OIL PUMP, STARTER, GENERATOR, BUT NO INLET OR EXHAUST DUCT

SFT-12

The gas generator section of the simple-cycle, free-turbine engine is nearly identical in design with that of the single-shaft engine and, hence, has a similar manufacturing cost. However, the addition of the free turbine adds significantly to the cost of the basic engine, since it doubles the turbine stages, main engine shafts, bearings, and output drives. Although the original power turbine as drawn was shrouded, vendor recommendations indicated a large potential cost saving ($15 to $20) in removing this shroud, and the costs as shown are for an unshrouded wheel. This change would affect efficiency adversely, with the actual magnitude unknown but expected to be approximately 1 to 2%.

In comparing the free-turbine and single-shaft versions of the simple-cycle engine, it appears to be far more expensive to supply the free-turbine engine, at an increment of $113 in direct manufacturing cost (10^6), than to use a more complicated transmission to condition the power output of the single-shaft engine for automotive drive. The more complex transmission is expected to require no more than an additional $50 (of direct cost) more than the less sophisticated transmission usable with the free turbine.

RSS-7

Comparison of the engine direct manufacturing costs for the 7:1 pressure ratio regenerated engine with those of the simple-cycle designs indicates that this regenerated design costs over 50% more than the single-shaft version (at equivalent rates of production). The regenerator, which is composed of metal screening matrices in a cast iron housing, results in slightly over 25% of the total direct cost. Materials in the turbine section were assumed identical with those in the simple-cycle engines, whereas the compressor impeller was assumed to be forged from steel in one piece. Steel was specified since the compressor exit temperature is in excess of that which would allow the use of an aluminum impeller.*

RFT-4

The manufacturing costs of the RFT-4 engine are the highest of the four types of engines considered in this study. The engine is considerably larger, heavier, and more complex than the other gas turbine designs, due to its low pressure ratio, low turbine net work per pound of airflow, need for a large regenerator disk, and the requirement for a free turbine with variable geometry. Essentially, it combines the cost increments of both the SFT-12 and the RSS-7 relative to the simple-cycle, single-shaft engine (SSS-12). Together, the regenerator and the power turbine are responsible for more than 50% of the basic engine cost. Of this total, the ceramic regenerator disk represents less than 8% of the cost, the primary cost in the regenerator being due to the seal system necessary to minimize gas leakage.

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* More recent stress calculations have indicated that the use of steel in the impeller would probably limit the pressure ratio to less than 6:1, so that a titanium impeller may be required for a 7:1 pressure-ratio engine. If titanium were specified, approximately $4 would be added to the direct manufacturing cost at the 10^6 annual quantity.

Comparison with Otto-Cycle Engine

A standard 300-cu in., V-8 engine with 1970 pollution-control equipment and standard accessories* is estimated to "cost" $200 to $250 on a comparable basis (total of direct manufacturing costs, as defined herein, at production rates in the order of 10^6 units annually).** It may be noted that if the Otto-cycle engine can be modified to reliably meet 1976 emission requirements, the direct manufacturing cost of the SSS-12 is predicted to be equal to it (following start-up and introductory expenses and at comparable production rates) if the direct manufacturing cost of these modifications is approximately $50 to $60, when the differences between the largest and smallest costs estimated for each engine are averaged.***

---

* Generator, starter, ignition system, radiator, fan, water pump, carburetor, fuel pump, oil pump.

** This type of information is, of course, very closely guarded by the major automobile manufacturers; also, for reasons discussed previously, the total of the direct-cost elements is not in itself significant. The estimate given above was pieced together from indirect information.

*** Of the two engines, the SSS-12 lends itself at least as readily to fully automated production techniques, thus minimizing the impact of future increases in labor rates.

CRITICAL MATERIALS

Requirements for Critical Materials

Because gas turbine engines are required to operate at high temperature for sustained periods of time, they require metals with critical-materials contents (i.e., tungsten, cobalt, nickel, and chromium) usually far in excess of the amounts normally used in conventional spark-ignition engines. In addition, stress requirements in the high-pressure-ratio (high-tip speed) engines dictate the use of titanium for the compressor rotors. While such materials are generally in short supply, on either a national or global scale, they have been specified and used in aircraft gas turbines because of their unique and valuable properties. The level of usage required is tolerable because of the relatively small number of aircraft gas turbines as compared with the automobile market. However, with a potential annual production of 10,000,000 automotive gas turbines, such as might exist if an alternative low-pollution engine were to replace the Otto-cycle automobile engine, an availability problem of quite a different magnitude may exist.

In assessing this problem, the use of these critical materials was specified as required for two of the gas turbines designed for this study, the SSS-12 and the RFT-4 (since these represent the smallest and the largest critical-material requirement, respectively), and the total amounts of these materials were summed for each engine. Then the national and global availability of each of the materials was analyzed and compared with the automotive requirements assuming an annual production level of 10,000,000 units (as called for in the study contract). In several cases, initial design specifications revealed potential supply problems, and suitable substitute materials having less critical availability were specified.

In this analysis, only the simple-cycle, single-shaft engine (SSS-12) and the conventional automobile gas turbine (RFT-4) were examined in detail. Furthermore, in this initial design iteration, the materials specifications may not be those ultimately used (nor necessarily correspond exactly to those presently anticipated in engines planned for production by the automotive industry), but they are consistent with good technical practice and low manufacturing cost.

The materials specified for these two engines are indicated in Tables X and XIII. The material compositions of major alloys are given in Table XVIII, and the critical-materials estimate is summarized in Table XIX. The amounts of all calculated critical materials are based only on the estimated weights of the component parts containing significant amounts of these elements; traces present in other parts were not included. No allowance is made for waste or excess materials, since it is assumed that the excess, as well as all chips produced, would be recovered and recycled. As noted in Table XIX, for all materials except titanium,

TABLE XVIII

CRITICAL MATERIAL CONTENT OF SELECTED ALLOYS

(In % Composition)

Material	Alloy						
	AMS 4928	Hastelloy-X	Udimet 700	IN 738	Inconel 706	Custom 455	IN 100
			% Composition				
1. Nickel	--	46.23	49.285	60.385	See Below	9.0	51.255
2. Cobalt	--	1.5	18.5	8.5	See Below	--	15.0
3. Nickel and Cobalt	--	--	--	--	41.0	--	--
4. Chromium	--	22.0	15.0	16.0	16.0	11.75	10.0
5. Titanium	89.2375	--	3.3	3.5	1.8	1.20	5.0
6. Manganese	--	1.0	0.15	0.20	0.35	0.5	--
7. Tungsten	--	0.6	--	2.6	--	--	--
8. Molybdenum	--	9.0	5.0	1.8	--	--	3.0
9. Zirconium	--	--	0.06	0.12	36.799	--	0.05
10. Columbium and Tantalum	--	--	--	2.7	2.9	0.3	--
11. Vanadium	4.0	--	--	--	--	--	10.0
12. Copper	--	--	0.1	--	0.3	2.25	--
13. Aluminum	6.1	--	4.3	3.5	0.4	--	5.5
14. Iron	0.3	18.5	4.0	0.20	0.006	74.47	--

Source: Selected vendor references plus 1971 Materials Selector, Materials Engineering, Mid-November 1970, Vol. 72, No. 6

TABLE XIX

ESTIMATE OF CRITICAL METALS CONTENTS IN SSS-12 and RFT-4

	Content per Engine (lb)	Consumption at Annual Production of 10^7 Units (tons)
SSS-12*		
Tungsten	0.14	685
Cobalt	0.43	2,155
Nickel	4.65	23,230
Chromium	1.54	7,695
Titanium	0.91	4,530
TOTAL	7.67	
RFT-4**		
Tungsten	0.15	750
Cobalt	2.03	10,145
Nickel	12.20	61,000
Chromium	2.84	14,200
TOTAL	17.22	

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* Critical metals are, primarily, in IN 706, Udimet 700, IN 738, Hastelloy X, and AMS 4928.

** Critical metals are, primarily, in IN 100, IN 738, and Hastelloy X.

the amounts of these critical materials would be greater in the RFT-4, with the difference, relative to the SSS-12, being as follows: tungsten, increase of 7%; cobalt, increase of 373%; nickel, increase of 162%; and chromium, increase of 182%. In addition to the unit values for each engine, the annual consumption of these materials for a production rate of 10,000,000 units per year is listed in Table XIX.

Availability of Critical Materials

Tables XX through XXIV have been prepared to show various statistics concerning the supply of each of the critical materials. In each case, the current US consumption is shown, together with both the current US and world production. Where US production is not adequate to its needs, the principal suppliers and the current import quantities are shown. A predicted future US consumption of the material is shown to lend insight into the relative future demand for the material exclusive of its demand for automobile gas turbines. Finally, the estimated world resources of each material are listed by principal source, including the US reserves even when they may be small compared with the principal sources.

For each of the critical materials, the annual needs listed in Table XIX for both the SSS-12 and RFT-4 gas turbines are related to the current level of US consumption and production to show the potential impact of automobile gas turbines on the use of these materials. The gas turbine needs are also related to future US needs of these materials for other than automobile gas turbines, and to the US reserves of each material.

With the use of the materials as specified and costed, the critical nature of the cobalt supply is seen to be the most severe (Table XX). The needs of the SSS-12 gas turbine would equal one-third the current US consumption rate, while the RFT-4 gas turbine would require 156% of the current US consumption rate. The normal growth of cobalt use is moderate so that even in the year 2000 the SSS-12 would require about one-eighth of all other uses, while the RFT-4 would need more than one-half that of all other applications combined. US dependence on cobalt imports is apparent in that its total reserves could supply only 2 year's requirements for the RFT-4 and 8 year's for the SSS-12, based on the calculated consumption rate of 10 million engines annually. The problem is even more critical in view of the lack of richness of US cobalt-containing ore. Obviously, continuing supply from Katanga and Belgian-controlled sources is important, both because of their extensive reserves and richness of ore.

TABLE XX

PRODUCTION AND CONSUMPTION OF COBALT

			Percent Needed for Automobile Gas Turbines(2)	
			SSS-12	RFT-4
Current Statistics(1) - tons				
Annual US Consumption	6,500	(30% World Prod.)	33	156
Annual US Production	-	-	-	-
Annual World Production	21,500		10	47
Principal Suppliers to US, Quantity - tons(1)				
Belgium - Luxembourg	1,700		-	-
Congo (Katanga)	1,320		-	-
Canada	520		-	-
France, Norway, and Other	1,070		-	-
Predicted US Consumption - tons (year)(3,4)	10,000	(1980)	22	101
	19,000	(2000)	11	54
Estimated World Resources(5) - tons				
Cuba	1,570,000	0.07-1% CO(6)	-	-
Congo (Katanga)	750,000	3-4 "	-	-
New Caledonia	440,000	0.2 -5 "	-	-
Northern Rhodesia	375,000	3-4 "	-	-
Canada	344,000	0.06-0.4 "	-	-
US	16,500	0.05-0.3 "	13	62
Remaining parts of Africa	13,000	1.6 "	-	-

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(1) Ref. 28

(2) Based on annual production of 10,000,000 units (Table XIX)

(3) Ref. 29

(4) Not including consumption in automobile gas turbines

(5) Ref. 30

(6) Cobalt content of ore

Despite the significant current use of nickel in this country (Table XXI), SSS-12 requirements would increase its use by 15% annually, and a problem might exist if the US were exclusively dependent on overseas supplies. Fortunately, Canadian sources appear secure and their reserves adequate to satisfy the demand for nickel for this engine. The RFT-4 poses a more different supply problem, however, because its requirements for nickel are more than two and one-half times higher. Its needs would require 38% of current US consumption and represent a significant fraction of the total US needs, even as far in the future as the year 2000. Its annual requirement for 12% of the US reserves, when added to the future demand for nickel for other uses, would put the US in short supply should Canadian sources ever be jeopardized.

Chromium occurs natually as chromite, which is composed of the oxides of chromium, iron, aluminum, and magnesium in varying quantities, all in solid solution. Since this ore contains these important commercial metals, it is usually shipped without separation of the ore into its separate component metals.

There are three principal industrial uses of chromite: metallurgical, refractory, and chemical. They account for 58%, 30%, and 12%, respectively, of the ore consumed in the US. Traditionally, the high-chromium chromite has been classified as the metallurgical grade, the high-aluminum as the refractory grade, and the high-iron as the chemical grade. It is the metallurgical-grade chromite which is usually converted to a ferro-chromite, which in turn is then consumed in the manufacture of alloy steels.

The data presented in Table XXII indicate that the chromium needs of automobile gas turbines represent no apparent problem, in view of the very small percentage of such needs relative to the current consumption for other uses, particularly in the case of the SSS-12. The RFT-4 requires more than twice as much chromium. The US dependence on the USSR and Turkey for high-chromium chromite would appear to represent a possible problem for such material, in general, but the small percentage required for the automotive gas turbine does not appear to add significantly to the existing problem.

The need for tungsten at a production rate of 10,000,000 SSS-12 gas turbines per year is equal to 12% of the current US consumption rate, and is equal to 13% of the annual US production rate (Table XXIII). With respect to the RFT-4, the problem of tungsten demand is approximately equal to the demand for the SSS-12. It should be noted that spark-ignition automotive engines currently employ 1/4 to 1/2 lb of tungsten; thus the use of 0.14 lb in simple-cycle gas turbines actually reduces by roughly one half the automotive quantity already contained in the US consumption value. Therefore, the substitution of 10,000,000 gas turbines per year for a like number of spark-ignition engines would subtract about 1/4 lb per engine from the demand for tungsten (about 1300 tons annually).

TABLE XXI

PRODUCTION AND CONSUMPTION OF NICKEL

		Percent Needed for Automobile Gas Turbines(2)	
		SSS-12	RFT-4
Current Statistics(1) - tons			
Annual US Consumption	159,300 (30% World Prod.)	15	38
Annual US Production	15,200 (3% World Prod.)	153	400
Annual World Production	528,600	4.4	12
Principal Suppliers to US, Quantity - tons(1)			
Canada	128,300	-	-
Norway	9,500	-	-
Other	3,000	-	-
Predicted US Consumption(3,4) - tons (year)	263,000 (1980)	8.9	23
	563,000 (2000)	4.1	11
Estimated World Reserves(5)			
Cuba	17,000,000	-	-
New Caledonia	16,000,000	-	-
Canada	6,000,000	-	-
Philippines	4,000,000	-	-
Puerto Rico	1,000,000	-	-
US	500,000	4.7	12

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(1) Ref. 28
(2) Based on annual production of 10,000,000 units (Table XIX)
(3) Ref. 29
(4) Not including consumption in automobile gas turbines
(5) Ref. 30

TABLE XXII

PRODUCTION AND CONSUMPTION OF CHROMIUM

		Percent Needed for Automobile Gas Turbines(2)	
		SSS-12	RFT-4
Current Statistics(1) - tons			
Annual US Consumption	400,000 (25% World Prod.)	1.9	3.6
Annual US Production	-	-	-
Annual World Production	1,607,000	0.5	0.9
Principal Suppliers to US, Quantity(1) - tons			
South Africa	131,000	-	-
USSR	125,000	-	-
Turkey	46,600	-	-
Philippines, So. Rhodesia, Others	39,000	-	-
Predicted US Consumption(1,3) - tons (year)	805,000 (1985)	1	1.8
Estimated World Reserves(4)			
South Africa	600,000,000 (5% High-Cr ore)	-	-
Southern Rhodesia	180,000,000 (50 ")	-	-
Turkey	3,000,000 (90 ")	-	-
USSR	3,000,000 (50 ")	-	-
US	2,400,000 (5 ")	0.3	0.6
Philippines	2,250,000 (20 ")	-	-
Other	7,650,000 (40 ")	-	-

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(1) Ref. 28

(2) Based on annual production of 10,000,000 units (Table XIX)

(3) Not including consumption in automobile gas turbines

(4) Ref. 29

TABLE XXIII

PRODUCTION AND CONSUMPTION OF TUNGSTEN

		Percent Needed for Automobile Gas Turbines(2)	
		SSS-12	RFT-4
Current Statistics(1) - tons			
Annual US Consumption	5,500 (16% World Prod.)	12	14
Annual US Production	5,100 (15% World Prod.)	13	15
Annual World Production	34,900	2.0	2.2
Principal Suppliers to US, Quantity - tons			
Canada	520	-	-
Peru	253	-	-
Australia	72	-	-
Other	67	-	-
Predicted US Consumption(1,3) - tons (year)	36,800 (1985)	1.9	2.0
Estimated World Reserves(4) - tons			
China (mainland)	1,066,000	-	-
US	71,400	1.0	1.1
Korea (North and South)	35,500	-	-
Boliva	43,600	-	-
Burma, Brazil, and Others	146,200	-	-

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(1) Ref. 28

(2) Based on annual production rate of 10,000,000 units (Table XIX)

(3) Not including consumption in automobile gas turbines

(4) Ref. 29

The growth of US tungsten production for applications other than in automotive gas turbines is shown to be very high, so that in 1985 the simple-cycle engines would represent only 1.9% of the country's needs. In these terms, it is apparent that the simple-cycle automotive gas turbine would add only a small burden to the demand in 1985. However, the overall supply problem is severe, since the US reserves are small relative to future needs, and the main source of supply (the Chinese mainland) may be unavailable.

Titanium occurs naturally as ilmenite ($FeTiO_3$) and rutile (TiO_2). It can also be consumed in the form of titanium slag (70% TiO_2) and titanium sponge (90+% Ti). Ilmenite is used primarily (99% of US consumption) as a pigment for paints, varnishes, paper, etc. Rutile is used in the production of welding rod coatings (50%), and in the production of titanium metal (43%). Twenty percent of the titanium consumed in the US is in the form of rutile.

The usage of titanium for the SSS-12 appears to pose no problems, since such usage represents less than 1% of the current US consumption (Table XXIV). Of course, the total US consumption figure noted includes all forms of titanium; hence the percentage needed by gas turbines should be multiplied by approximately 5.0 to account for the fact that only one-fifth of that consumed is in the form of rutile, thereby resulting in a rutile percentage of just over 3% of current consumption. In terms of the US reserves, the annual SSS-12 needs are about 0.25% after adjustment is made for the low percentage of the reserve that exists in the form of rutile. The RFT-4 requires no titanium.

Evaluation

Based on the above estimates, only cobalt and nickel can be considered critical in terms of the needs of automobile gas turbines relative to materials availability. The use of both metals is a result of the requirements for refractory alloys for the hot parts of the engine, and to a large extent, they are interchangeable constituents as price (criticality) varies. Since no one engine model or manufacturer will produce all the engines manufactured in the US, it is obvious that many similar materials which differ in specific alloy content will be used. In general, it can be stated that a far greater problem would exist for the RFT-4 than for the SSS-12 type of engine, but that even in this case the total requirement for nickel, if no cobalt were used, might not represent a highly significant obstacle, and the interchangeability of nickel and cobalt/tungsten specifications offers many desirable tradeoffs from a price and critical-materials standpoint should any of these metals be in short supply.

TABLE XXIV

PRODUCTION AND CONSUMPTION OF TITANIUM

		Percent Needed for Automobile Gas Turbines(2)	
		SSS-12	RFT-4
Current Statistics(1) - tons			
Annual US Consumption(3)	473,000 (39% World Prod.)	1.0	-
Annual US Production(4)	303,400 (30%(5) ")	1.5	-
Annual World Production(5)	1,230,000 -	0.4	-
Principal Suppliers to US(1), Quantity - tons			
Australia	117,400	-	-
Canada	63,500	-	-
Sierra Leone, Others	1,500	-	-
Predicted US Consumption(1,7) - tons (year)	900,000 (1985)	0.5	-
Estimated World Reserves(8) - tons			
US	33,000,000 (4%(9))	< 0.1	-
Canada	30,000,000 (not indicated)	-	-
Other	60,000,000 (not indicated)	-	-

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(1) Ref. 28

(2) Based on annual production rate of 10,000,000 units (Table XIX)

(3) Includes ilmenite, rutile, titanium slag, and titanium sponge consumption

(4) Contains only information for ilmenite; rutile information "withheld as proprietary"

(5) US ilmenite production percentage of world ilmenite production

(6) Includes both ilmenite and rutile

(7) Ilmenite and Ti slag only; not including consumption in automobile gas turbines

(8) Ref. 29

(9) Percent of titanium reserve as rutile

There are several additional considerations regarding the criticality of the nickel supply with regard to the automobile gas turbine application. Approximately one-half pound of nickel goes into present automobile engines, so that the total additional requirements for automobile gas turbines would be less than that indicated above. Furthermore, this level of consumption will increase due to increased use of high-temperature materials to meet emission requirements for piston engines. Also, the heavy concentration of nickel in critical parts of the gas turbine engines makes reclamation an attractive procedure. Thus, as scrappage rate increases with time, more of the total nickel requirement could be met with reclaimed metal should the situation become critical enough to justify this process economically. Finally, the eventual use of ceramic parts and ceramic coatings in future gas turbines for hot-end parts such as turbine nozzles, flame tubes, etc., could significantly reduce critical-materials requirements as well as lower costs.

In view of the above analysis and qualifying considerations, the widespread adoption of automobile gas turbines, similar in type to the SSS-12 studied here, is not constrained by the availability of projected supplies of critical materials.

ANALYSIS OF OWNERSHIP COSTS

Retail Price to the Consumer

While the objective of this study was to estimate engine manufacturing costs, the ultimate customer (i.e., the automobile buyer) is concerned with the retail price of the entire automobile. A detailed analysis was beyond the scope of this study, but rough estimates have been prepared for comparing retail prices of automobiles powered by each of the four study engines, as well as automobiles powered by 1970 and 1976 Otto-cycle engines. These comparisons are based on six-passenger, standard-size sedans, and are shown in Table XXV. It must be emphasized that these estimates are made for illustrative purposes only, since they are not based on detailed information regarding pricing policies within the automotive industry. In this example, the following considerations apply, and the breakdown in Table XXV is intended to facilitate the substitution of different assumed values should other than a single-point comparison be desired.

The comparison is prepared to be consistent with a US Department of Transportation report entitled "Cost of Operating an Automobile" (Ref. 31). Thus, the Ref. 31 purchase price of $3374 for the 1970 Otto-cycle automobile is selected as a base value as presented in Table XXV. This price includes V-8 engine, automatic transmission, power steering, and Federal excise tax, and includes an unspecified discount from the manufacturer's suggested list price.

The retail prices which are shown in Table XXV are estimated by dividing the automobile price into two elements. The first element is that portion of the price representing the engine, and the second element represents the entire remainder of the vehicle.

The starting point for this analysis is the vehicle adjusted curb weight, W_7* (superscript * is used for all adjusted weights), from Table II, and these figures are entered on the first line of Table XXV. Since W_7* includes the weights of those fluids (water, fuel, etc.) which do not determine the vehicle retail price, the excess weight must be subtracted from the curb weight to yield a shipping weight, which is presumed to be the basis for establishing the retail price. These fluid weights are estimated as W_{10}, and the corresponding shipping weights shown as W_{11}*. The engine weights (W_1, Table II) are then subtracted to establish a residual vehicle weight, W_{12}*.

The portions of the retail price due to the engine and the residual vehicle are then estimated as follows. In the case of the OC-70 engine, the unit equipped-vehicle price is $3374 ÷ 3800 = 89¢/lb. While it is expected that an OC engine is likely to have a higher unit price than the rest of the vehicle, the

TABLE XXV

ESTIMATED AUTOMOBILE RETAIL PURCHASE PRICES

	Computation	OC-70	OC-76	SSS-12	SFT-10	RSS-7	RFT-4
W_7* - Adjusted Curb Weight - lb	$W_3 + W_4* + W_5$	4000	4140	3420	3410	3490	3710
W_{10}* - Estimated Weight of Fluids - lb	$W_7* - W_{10}$	200(1)	215	200	200	145	160
W_{11}* - Shipping Weight - lb		3800	3925	3220	3210	3345	3550
W_{12}* - Residual Vehicle Weight - lb	$W_{11}* - W_1$	3135(2)	3200	3060	3010	3055	3060
Portion of Price Due to Engines - $		590(2)	745(3)	740(4)	1150(5)	1200(4)	1720(4)
Portion of Price Due to Residual Vehicle - $		2780	2850	2720	2680	2720	2720
Estimated Vehicle Average Retail Price - $		3374	3595	3460	3830	3920	4440
Δ Retail Price Relative to OC-70 - $			+225	+ 86	+366	+546	+1066

(1) 35 lb coolant, 125 lb fuel, 10 lb crankcase oil, 30 lb miscellaneous = 200
(2) At 89¢/lb
(3) Includes $100 additional for pollution control devices
(4) Direct manufacturing cost x 2.7
(5) Direct manufacturing cost x 2.7+ $100 for engine braking device

engine portion is conservatively estimated at 89¢/lb for a 665-lb (dry) engine. For the OC-76 engine, the price is also estimated at 89¢/lb (725 lb dry), but an additional $100 is added as an allowance for the additional costs of pollution control devices (which might be catalysts, reactors, fuel injection systems, etc.). All of the gas turbine engines have retail prices estimated by multiplying their direct manufacturing costs by 2.7, except that the SFT-12 engine has an additional $100 allowance to provide for an additional device (variable power turbine geometry or power transfer, including control functions) to provide satisfactory engine braking.

The factor of 2.7 can be rationalized as follows. A lumped burden of 130% can be applied to the total of direct manufacturing cost elements identified herein to estimate an approximate total manufacturer's price (dealer's cost); for the typical full-size "standard" automobile considered, the dealer's markup to sticker price is probably approximately 23%, and a typical discount to an actual sale price on a "clean deal" (or as an exaggerated trade-in allowance) is probably about 10%. Thus,

$$(1 + 1.3) \frac{1}{1-0.23} (1 - 0.10) = 2.7$$

The portion of the retail price due to the remainder of the vehicle (without gas turbine engine) is arrived at by multiplying W_{12}^*, the residual vehicle weight, by 89¢/lb. The price of the automobile is then arrived at by summing the engine and residual vehicle portions of the price.

The last line on Table XXV shows the differences in retail price relative to the present OC-70 engined automobile, using the above assumptions. It is seen that the OC-76 automobile is estimated to cost $225 more than the OC-70 automobile; this estimate is at the low end of the $200-to-$500 range frequently mentioned by various knowledgeable sources. Of the four gas turbines studied, only the SSS-12 engined automobile is lower than this figure, and it is encouragingly close (within less than $100) to present experience as represented by the OC-70 automobile. The comparison between the single-shaft and free-turbine versions of the same engine (SSS-12 and SFT-12), with a price difference of $280 in favor of the SSS-12, appears to clearly resolve the issue in favor of a single-shaft engine, even though it requires a more complex and costly transmission. (As mentioned elsewhere in this report, further studies and actual demonstrations of transmissions suitable for single-shaft engines are required to substantiate this finding.)

On the basis of price, the RSS-7 engined automobile is at a severe disadvantage ($546 above the OC-70 automobile), but as will shortly be discussed its outstanding fuel economy tends to compensate for this initial disadvantage. The

RFT-4 would appear to be excluded as a feasible alternative engine on the basis of cost alone (> $1000 increase over present automobiles) unless drastic reductions can be achieved in its probable manufacturing costs.

It must be emphasized that the relative price differences shown for the 1976 automobiles could change considerably. The total $139 differential between the SSS-12 engined automobile and the OC-76 engined automobile could easily rise to $300 should the retail cost of the pollution control devices for the 1976 Otto-cycle automobile increase by an additional $80, and should the direct manufacturing cost of the SSS-12 engine decrease by $30 ($81 retail price). In addition, significant savings might be available in the gas turbine car from a direct air conditioning system integrated with the simple-cycle engine relative to the present pump and condenser cycle now used; this comparison becomes meaningful as most cars become air conditioned. On the other hand, it is possible that the uncertainties associated with the manufacturing cost of the simple-cycle engine and its transmission could cause its price to increase to the point where the simple-cycle gas turbine automobile would cost more to manufacture than the 1976 Otto-cycle automobile assumed above, especially if a major breakthrough occurred which permitted the future Otto-cycle automobile to meet emissions standards with little or no cost penalty.

Obviously, there are many uncertainties inherent in the comparisons shown in Table XXV because of the many assumptions involved. For example, it can be argued that the prices of all the 1976 vehicles should rise more than indicated because of the additional costs involved in research and development for low-emissions engines, whether Otto-cycle or gas turbine. On the other hand, the public demand for "a brand new concept" turbine-engined automobile might also increase car sales considerably, as innovations in automobiles have done in the past (Ref. 32), even if their retail prices were slightly higher. Despite these uncertainties, the predominant indication is that the simple-cycle gas turbine can indeed be manufactured at a cost which allows a retail selling price which is competitive with a pollution-controlled Otto-cycle engined automobile.

The weights and costs shown in Tables II and XXV do not include additional weights and costs of meeting future mandatory Federal safety regulations. These considerations are excluded from the analysis in order to avoid complicating the simplified comparison shown. This procedure is justified because these safety regulations will probably have second-order effects on the propulsion system comparisons shown. In general, however, it should be noted that the major technical difficulty in achieving acceptable crash-proof cars, i.e., weight problems, can be at least partially alleviated by a lighter-weight power plant, such as the automobile gas turbine engines studied; also, the small size of the gas turbine engine permits a high degree of flexibility in location and mounting arrangements, thus facilitating design of the vehicle for safety.

Depreciation and Operating Costs

An ownership cost comparison may be derived by using the estimated retail prices in the previous section, employing the fuel economy comparisons derived from the automobile computer performance program previously described, and making first-order maintenance cost estimates. Use has also been made of the report "Cost of Operating an Automobile" issued by the US Department of Transportation, Federal Highway Administration, Bureau of Public Roads, February 1970 (Ref. 31). The engine-related operating costs appear to be, in order of importance: the depreciation cost of the original vehicle, the fuel consumption, the maintenance, and the oil consumption. The following assumptions and/or considerations were made:

1. A single-point comparison for an average driver operating as described in Ref. 31 provides a first-order estimate of operating cost comparisons.
2. The comparisons pertain to standard, six-passenger, four-door sedans.
3. The retail price of the vehicles compared is taken from Table XXV. Depreciation costs for all vehicles are then calculated on a fixed-percentage basis of 73% for the first four years and 100% for the full ten years, as given in Ref. 31.
4. The cost-of-ownership analysis is most significant at the 4-year (assumed original owner) and 10-year (assumed vehicle life) periods, and the analysis is therefore limited to these points.
5. The maintenance cost breakdowns used in these comparisons involved three separate assumptions as follows:
    a. The maintenance cost for the propulsion system of the OC-70 automobile is assumed to be in proportion to the weight ratio of the propulsion system to the entire vehicle, i.e., W_P/W_C = 1300/4000 (corresponding to the vehicle weight breakdown W_3/W_7 from Table II).

b. A frequency of repair for the gas turbine engines is assumed to be one-half that of the OC-70 engine*, and the OC-76 engine is assumed to require repairs (or adjustments) 33% more frequently than the OC-70 engine**.

c. The cost of each repair is assumed to be in direct proportion to that portion of the retail price which is attributable to the engine (from line entitled "Portion of Price Due to Engines," Table XXV).

6. Fuel economy comparisons were derived by decreasing the resistance equation values (Eqs. 22 and 23) by 13% for input to the UARL Typical Driving Cycle (Fig. 25), which is typical of country driving, in order that the OC-70 automobile would deliver the 13.8 mpg set in the reference report (page 9, Ref. 31). These same inputs were then used with the fuel consumption input curves (Figs. 27 and 28) to estimate the fuel consumption of the gas turbine engines on the identical driving cycle. In the case of the OC-76 automobile, an overall 10% efficiency penalty relative to the OC-70 is applied***. This penalty accounts for a 5% decrease due to reducing compression ratio and spark retardation, and a 5% decrease due to exhaust gas recirculation.

7. Fuel costs were estimated at 35¢/gal for the OC-70 (as per Ref. 29), 33¢/gal for the gas turbine engines (based on an East Hartford area survey which showed an average differential of 2¢/gal between diesel fuels and regular gasoline), and 36¢/gal for the OC-76 (assuming 1¢/gal is the retail price differential to the consumer for unleaded gasoline).

8. Oil costs, although a minor item, are obviously engine-related. They are included at the following rates: OC-70 and OC-76, $15/year as per Ref. 31; all gas turbine engines, $4/year, which includes makeup oil and 2 changes, at $2/quart.

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* A decrease appears justified on the basis of the fewer tuneups, adjustments, etc., required; however, the 50% magnitude of this reduction is an unsupported assumption at this point in the development cycle.

** An increase appears justified on the grounds of manufacturer's maintenance recommendations to insure meeting emissions requirements. The magnitude of the increase assumed here (33%) is an unsupported assumption.

*** The 10% penalty assumed here is optimistic in view of current automobile industry projections. For example, one "maximum effort" system resulted in a 25% economy loss over the FDC while average NO_x levels still exceeded the allowable by a factor of two (Ref. 33).

Table XXVI summarizes the results of this cursory ownership cost analysis. As can be seen, the changes relative to the OC-70 automobile reflect engine-related operating cost penalties ranging from -7% to +14%, with the RSS-7 automobile showing the most favorable comparison (at the 10-year point) and the RFT-4 the least favorable (at the 4-year point). The last line of Table XVI shows whether or not the predicted costs are within the OAP requirement not to exceed 10% of present experience as established by Ref. 11. These results are also graphically displayed in Fig. 52, where the oil and maintenance costs are combined for simplicity. The broad conclusion which can be drawn is that it appears that the SSS-12 and the RSS-7 engined automobiles would be at least competitive in ownership costs with present experience (OC-70), while the RFT-4 and the OC-76 are at least 10% higher than the present ownership cost experience.

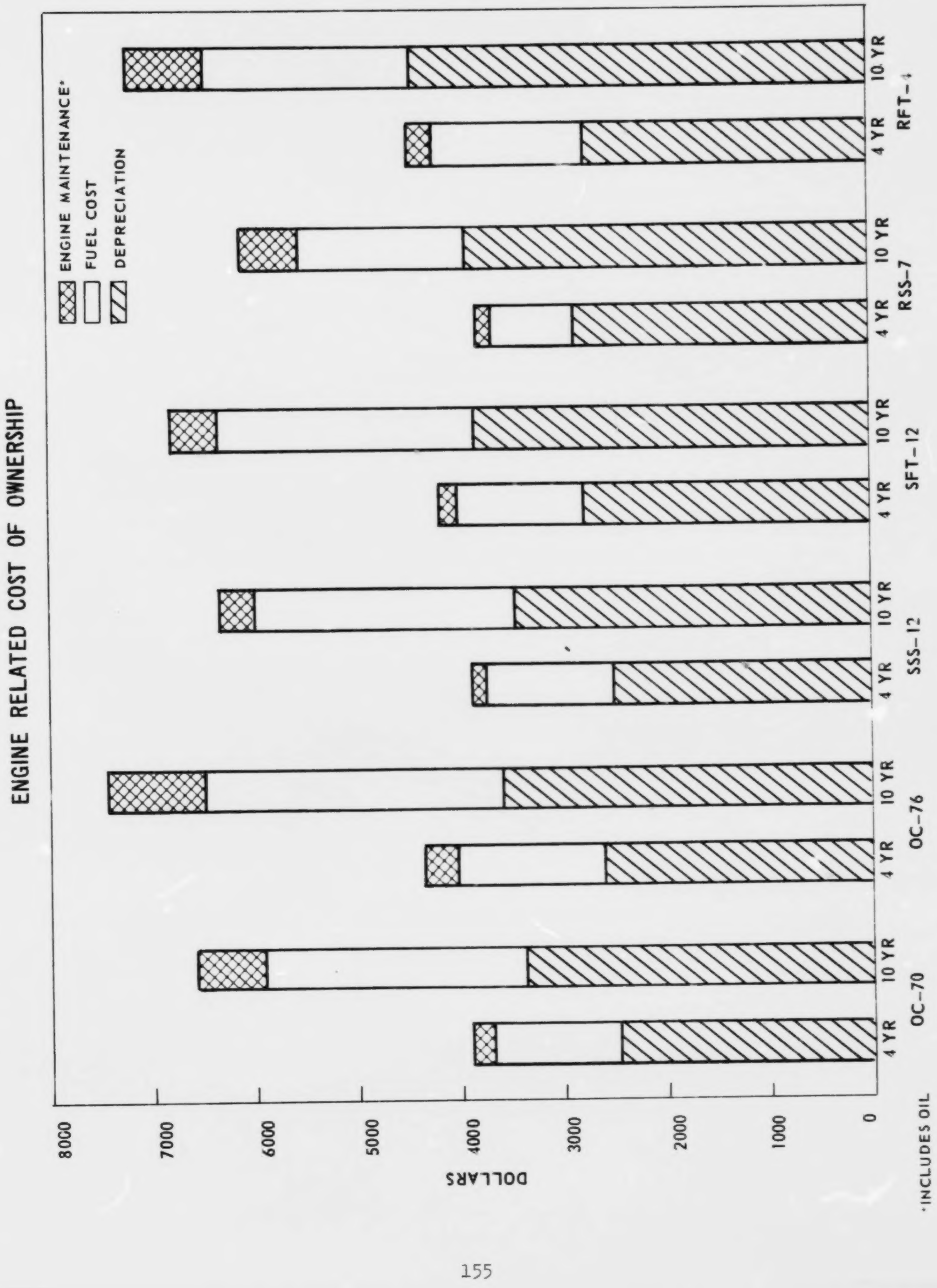
Since it has been generally believed that the fuel economy of a simple-cycle engine would be unacceptable for automobile usage, the prediction that SSS-12 fuel costs are practically identical to present experience for the typical country driving cycle assumed is of particular interest. Several observations regarding comparative fuel cost predictions on urban cycles are also in order, since the SSS-12 is operating under its least competitive fuel flows on these typically low power cycles. Using the FDC fuel economy predictions (Table VII) as representative of urban driving, and the same fuel price assumptions as were used above, the fuel cost of the SSS-12 would exceed that of the OC-70 by 19% on the FDC. However, since the power requirements for this cycle are so low (about 20 hp), the difference in actual fuel cost is only 15¢/hour (89¢/hour versus 74¢/hour). Furthermore, as mentioned on p. 81, optimization studies are currently under way to reduce part-load fuel consumption of the SSS-12. Another promising technique under study is to reduce idle fuel flow by reducing the arbitrarily selected 70% idle speed. While this procedure is at the expense of response time, this effect can be compensated for by permitting adjustable operator selection of idle speeds from maximum economy (and minimum emissions) to maximum performance. The combined effect of optimization and idle speed selection can probably reduce the 19% fuel cost differential to less than 10%. Finally, it should be noted that if a 25% decrease in fuel economy is incurred by the OC-76, as per Ref. 33, the SSS-12, as designed, offers better fuel cost over the FDC than the future pollution-treated Otto-cycle engine.

TABLE XXVI

OWNERSHIP COST ANALYSIS

	Engine											
	OC-70		OC-76		SSS-12		SFT-12		RSS-7		RFT-4	
	4 yr	10 yr	4 yr	10 yr	4 yr	10 yr	4 yr	10 yr	4 yr	10 yr	4 yr	10 yr
Depreciation - $	2460	3374	2620	3595	2520	3460	2795	3830	2860	3920	3250	4440
Fuel Economy (mpg)	13.8	13.8	12.5	12.5	13.4	13.4	13.4	13.4	20.6	20.6	16.8	16.8
Fuel Cost - $	1239	2531	1402	2863	1230	2513	1230	2513	798	1629	983	2008
Oil Cost - $	63	161	63	161	16	40	16	40	16	40	16	40
Engine Maintenance - $	168	494	279	820	104	313	147	441	170	512	240	720
Total Engine Related Costs - $	3930	6560	4364	7439	3870	6326	4188	6024	3844	6101	4479	7208
Change Relative OC-70 - $	---	---	11%	13%	-1.5%	-3.5%	+6.6%	4.0%	-2.2%	-7.0%	+14%	+10%
Within OAP goals for ownership cost (+10%)	---	---	No	No	Yes	Yes	Yes	Yes	Yes	Yes	No	Yes

FIG. 52



ENGINE RELATED COST OF OWNERSHIP

*INCLUDES OIL

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RESEARCH AND DEVELOPMENT REQUIREMENTS

Based on the results of this study, an appropriate development program in gas turbine technology and manufacturing technology is recommended. This program assumes that the major interest is in the simple-cycle, single-shaft, high-pressure-ratio engine (SSS-12), since it appears to have the lowest manufacturing cost potential and the best potential to meet NO_x emissions requirements, while maintaining acceptable operating costs. The purpose of this development program is to demonstrate the feasibility of this approach within a time span which allows for initial engine production during 1976.

The program, as shown in Fig. 53, is divided into two phases. The first phase would be devoted to concept validation, basic component research, fabrication, and test of several demonstrator engines; development and demonstration of suitable transmissions; and preliminary vehicle demonstration. This phase is assumed to begin during 1971 and would encompass about two and one-half years. Phase II, which would follow, would include the design, development, and test of prototype engines, as well as a more extensive vehicle demonstration (fleet-test) program, and would require an additional 2 to 2-1/2 years.

Phase I Program

The Phase I program would consist of three parts. The first part would encompass an engine optimization study, a transmission feasibility study, and research, development, and demonstration of the following engine components: combustor, compressor, turbine, gear box, bearings, and fuel control; also it would include a feasibility demonstration of high-ductility forging for this application. The second part would consist of engine and transmission design, fabrication, and test. The third part of the Phase I effort would be the demonstration of engines, with two or three suitable transmissions, in automobiles. The first two parts can and should be accomplished concurrently if initial production is to occur in 1976. Each task is discussed below.

The Phase I program is envisioned as being at least partially funded by OAP in accordance with its desires to devote research effort to achieve practical solutions to gas turbine products in order to stimulate the adoption of this technology in the automobile industry.

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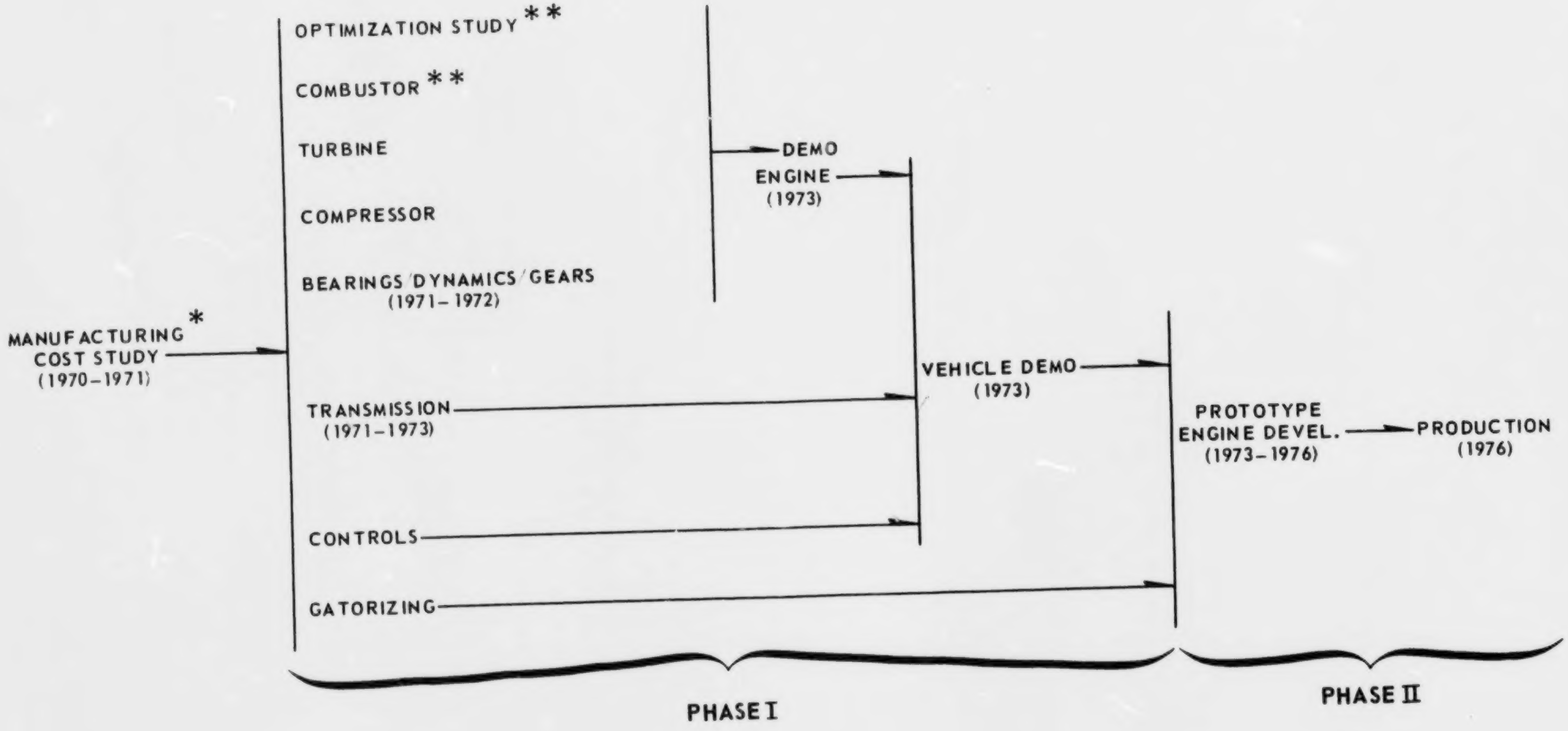
FIG. 53

RECOMMENDED DEVELOPMENT PROGRAM


MANUFACTURING COST STUDY (1970–1971) *
OPTIMIZATION STUDY **
COMBUSTOR **
TURBINE
COMPRESSOR
BEARINGS/DYNAMICS/GEARS (1971–1972)
TRANSMISSION (1971–1973)
CONTROLS
GATORIZING
DEMO ENGINE (1973)
VEHICLE DEMO (1973)
PROTOTYPE ENGINE DEVEL. (1973–1976)
PRODUCTION (1976)
PHASE I
PHASE II

*OAP SPONSORED STUDY COMPLETED

**OAP SPONSORED STUDY IN PROGRESS

Combustors

Parallel studies are now under way under OAP sponsorship* to develop and demonstrate suitably sized combustors designed from conception for low emissions for both regenerative and advanced simple-cycle gas turbine engines. The combustors developed under this program should be directly transferable to the design of automobile gas turbines, including the simple-cycle engine. It is expected that this study will demonstrate that the advanced simple-cycle engine will meet or surpass the 1976 Federal standards for NO_x (and other species) production**.

Turbine

Hot and cold testing of the radial turbine specified for the engine, at the correct Reynolds number and correct size, will be essential to demonstrate the high work and high efficiency potential of this component. The program should include the exhaust diffuser, since its performance is critical in determining overall turbine performance. The turbine test program would also allow for assessment of performance penalties, if any, due to modifications for low-cost production or durability such as thickened blades, reduced number of nozzles, and two-part construction.

Compressor

Test results presently being recorded at UACL lead to high confidence in the compressor performance projected for the SSS-12 engine. However, further rig demonstration of compressor capability at near-correct Reynolds numbers, size, and speed would appear to be very worth while, particularly for extensive investigation of inlet guide vane variability effects. Not only would this program validate the actual performance of the compressor to be used in the engine, but it could also be used to explore the performance losses, if any, involved with tradeoffs for lower-cost mass-production manufacturing techniques, such as increased clearances or thickened leading edges.

Bearings/Dynamics/Gears

Because of its very high input rotational speed and the need to minimize cost, inertia, and idling horsepower loss, the design of the gearbox represents a high state-of-the-art achievement, which will require development. This gearbox will be easier in other respects than many other gas turbine applications, since it is cold and employs very few meshes. A demonstrator rig program will both validate the feasibility of the concept and will save development time and money.

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* In response to RFP EHSD 71-NEG.-104, issued by EPA in December 1970.

** The study goal is to demonstrate compliance with the standards for both the regenerative and simple-cycle combustors. This discussion deals only with a development program for the simple-cycle engine.

The bearings proposed for the engine represent an application at the extreme limit of the current state of the art because of the extremely high rotational speeds of the shaft and the requirement for low-cost production. It will be necessary to develop the bearings for this application in order to demonstrate their feasibility. The reference approach should be rolling-contact bearings, although a parallel effort on air bearings may be worth while in the long run.

Fuel Control

The fuel control accounts for a significant portion of the manufacturing cost of the engine. A development program is required for a suitable low-cost control which includes a proper assessment of performance losses expected as a function of reduced manufacturing cost and function complexity. Inasmuch as fuel-control requirements will not be sufficiently well known until some of the other studies are performed, the fuel-control study should start later than the others.

Transmission

Transmissions suitable for the SSS-12 engine have not been studied in a degree of detail comparable to that of the engines. While it is believed that suitable transmissions for single-shaft engines can be evolved, and that they may not be more expensive nor more complex than automatic transmissions presently used in present-day automobiles, it is necessary to examine promising candidates from the standpoint of feasibility, and to design, fabricate, and test two or three suitable transmissions in order to validate this contention and insure validity of the single-shaft concept.

The transmission program is envisioned as being divided into three parts. The first part would be a six-month feasibility study in which all candidate systems would be studied and evaluated, and two or three of the most promising candidates selected for further development effort. The second part would be the design, fabrication, and test of the two or three most promising approaches. Dynamic calculations and simulations of the transmission, engine, and fuel control interactions should be performed to insure stable and adequate performance under all conditions. The third part of the study would be actual demonstration of the selected transmissions in automobiles with the demonstrator engines. At the conclusion of the program, it would be demonstrated that one or more suitable transmissions exist for the single-shaft engine. The fabricated transmissions would then be available for use during Phase II of the engine development program.

High-Ductility Forging Techniques

A key manufacturing technique for reduction of manufacturing costs of simple-cycle gas turbines to a level which will be competitive with present spark-ignition engines is the use of high-ductility forging techniques which are especially appropriate for the refractory materials and titanium used in the turbine and compressor rotors of these engines. The resulting components would have production costs normally associated with cast rotors, yet have the strength and structural integrity associated with forged rotors. Although this technique will probably prove to be extremely cost-effective in the production of these parts in the future, investigations of the feasibility of this method toward automated production of complete rotors for automobile applications have not yet been made. A program is therefore suggested which will investigate the feasibility of producing titanium compressor rotors and refractory-alloy metal turbine rotors using high-ductility forging. The program would end with a demonstration of techniques which would be adaptable to high-volume mass-production methods. Testing of the resultant components could be accomplished in the compressor and turbine programs outlined above.

Demonstrator Engines

Once component efficiencies are demonstrated unambiguously, and components developed on the rigs mentioned above, an attractive way of demonstrating the basic feasibility of the entire engine is to combine the compressor, combustor, turbine, gearbox, and bearings in an actual engine demonstration. The demonstrator engine concept minimizes cost by concentrating on the key engine elements, while controls, lubrication, power takeoff, and all accessories can be handled in a breadboard fashion, without final definition of produceability. Up to one year of stand testing is normally required to develop the demonstrator engines to the required standards of reliability and performance for a meaningful vehicle demonstration.

Vehicle Demonstration

At least two automobiles powered by demonstration engines and differing in transmission and installation details should be prepared for testing. The testing will define vehicle handling, acceleration performance, drivability, and customer reactions, as well as permit emissions measurements under prescribed driving conditions. Feedback from this program will be essential to define the specifications for Phase II which follows.

Phase II Program

The Phase II program involves three main parts. The first part -- that of the production prototype engine development -- presumably could be wholly or partially funded by OAP, depending on circumstances which will more clearly be defined by the time of inception of Phase II. The second part of the Phase II program -- the extensive fleet testing of the resulting gas turbine-powered vehicles -- could be performed by various Government agencies under OAP supervision, similar to programs which have become part of the Federal Clean Car Incentive (FCCI) Program. This procedure would materially reduce the cost to the prospective manufacturer of the extensive and otherwise expensive fleet testing required to prove the engine and identify problems which are visible only under these conditions. The final part of the Phase II effort -- production planning -- is envisioned to be the problem of the eventual manufacturer(s).

Detailed discussion of the Phase II program and its timing depends to a large degree upon the results and reception of the Phase I demonstrator program, and is therefore premature at this time.

Expected Development Program Results

Should the above program be adopted by OAP, the following results should be reasonable expectations:

1. Assuming that significant portions of this program commence during 1971, the key components should be running on rigs during 1972, and their feasibility should be largely demonstrated in that year. These components would include the combustor, the compressor, the turbine, the bearings, and at least one transmission.

2. First run of a demonstrator engine on a test stand will be late in 1972.

3. Vehicle testing with demonstrator engines will commence in late 1973, and will continue until testing of production prototype-engined vehicles commences.

4. Design, fabrication, and first run of production prototype engines will occur in 1974, with extensive vehicle testing commencing in 1975 and continuing into 1976.

5. Engines could be placed in initial volume production (50,000 to 250,000 annually) some time between mid-1976 and mid-1978, depending upon conditions which are discussed below.

Detailed analysis concerning the time span required to place the SSS-12 engine in full production was beyond the scope of this study. Several comments regarding pertinent aspects of this program appear to be in order, however.

First of all, it is important to note that under normal conditions significant differences exist between the time a product is introduced to the automobile market and the time when it penetrates a majority of the market, even when the new product has a clearly superior potential. The reasons for this phenomenon are many and varied, but obviously include such diverse variables as consumer desires, marketing strategies, profit maximization, risk minimization, price/cost relationships, introduction problems (such as unanticipated problems experienced with early production models, and the consequent recall expenses and embarrassment to the manufacturer), and a factor which has recently become important, i.e., legislative deadlines. Historical data concerning the rate of adoption of major innovations in the automobile industry (automatic transmissions, air conditioning, power brakes, power steering, short-stroke overhead valve V-8's, etc.) typically indicate spans of from 10 to 20 years from introduction before significant penetration (say, 2/3 to 4/5) of the annual market is reached.

Thus, if the marketing situation which has existed in the past were to continue in the future, it could be speculated that an evolutionary market penetration of a successful gas turbine engine might reach 2/3 to 4/5 of the market in, say, 10 to 15 years, regardless of which of the designs considered in this study were selected; all of them were assessed to be currently in a technological development stage sufficient to allow initial limited production* in 1976-78, provided an appropriate development effort, such as the one described above, were launched with this goal in mind**. The 1978 date represents an effort which is in accord with a normal 4-year period to place fleet-tested hardware into production, while the 1976 date would require a more intensive development effort, some redundancy in production tooling orders, a considerable compression of the 4 years time from production go-ahead to actual production, and possibly some compromise with performance. Nevertheless, there is no reason to assume at this time that the 1976 date for limited production is unrealistic, although obviously more study is needed before definitive statements regarding this prospect can be made. Under these conditions, initial market offerings would be limited to one or a select few of a manufacturer's total line, with additional lines added as time progressed.

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* For purposes of this discussion, limited production is assumed to mean 50,000 to 250,000 units annually.

** Since the regenerator designed for the RSS-7 engine would require additional rig-testing to adequately develop it, the RSS-7 would need to be redesigned with a ceramic regenerator to meet this date.

However, historical rates of adoption of innovations, marketplace decisions, and normal business processes will obviously not be the determining factors concerning the rate of adoption of low-emissions engines. The Administrator of the Environmental Protection Agency has stated that the Federal Government intends to fully enforce strict anti-pollution standards, even if it means Americans will have to pay more for cars that may run less efficiently (Ref. 34). Since the standards are expected to apply to 1976 model-year autos, the clear implication is that these standards must be met in September of 1975, if at all possible. If it should be demonstrated that the SSS-12 engine (or any other new type of low-emission engine) offers the best means of achieving this requirement, the sudden dislocation of producing up to 10 million vehicles annually with completely new types of engines and transmissions would require a massive effort, whose dimensions have not been seriously studied to date.* Any investigation of the impact of this type of implementation would necessarily have to consider a further compression of the engine development time scale described above, and the additional effort and costs required to do so.

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* A study of the impact of low-emission alternatives to conventional automobile engines is currently under way under DOT sponsorship (DOT-OS-10200 "Study of Mass-Produced Advanced Automotive Power Systems"). The prime contractor is International Research & Technology Corporation, with subcontract participation by the United Aircraft Research Laboratories.

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APPENDIX I

COMPUTER PROGRAM USER'S GUIDE

Program input consists of cards arranged in groups of 2 or 3, each group containing input for a "performance interval", and the run consisting of as many performance intervals as desired. A performance interval is the computer simulation of a selected performance mode option, such as constant acceleration, constant horsepower, or a particular duty cycle. For each performance interval, a vehicle must be defined in terms of weight, frontal area, drag coefficient, etc. In addition, interval data, which comprise information such as road grade, adhesion coefficient, and printing time interval, must be specified.

Input Cards

There are three types of input cards: (1) Vehicle cards, (2) Control cards, and (3) (Performance) Interval cards. The layout of the input on each card is illustrated in Fig. 54 which gives the column spacing, the Fortran symbols, and the definition for each input variable.

(1) Vehicle card - This card contains the parameters which physically describe the vehicle, such as weight, frontal area, etc. Most of the vehicle parameters which appear in Table IV also appear on this card, and no further discussion of them will be given. The variables in columns 41-60, however, require some explanation. The variable HPMAX defines maximum engine horsepower and is used to compute normalized horsepower. The variable VMAX specifies the vehicle design speed at which 100% transmission output speed occurs. It does not constrain vehicle performance to that speed, but rather represents a short cut in an iterative process which would be necessary to determine exact vehicle top speed. VMAX is used to compute percent transmission output speed at any given vehicle speed. The variables ENTYPE and TRTYPE specify the particular engine and transmission in the vehicle, and their assigned values are used by the computer to perform table lookups for sfc, etc., in the proper location in the tabular data storage array (described later). The actual number values for ENTYPE (1.0 to 4.0) and TRTYPE (1.0 to 4.0) correspond to the computer-code numbers used in Table V for engine and transmission types.

The format for this card is 16F5.1. This means that each input variable (including its decimal point) may occupy any or all of the five spaces assigned to it as shown in Fig. 54.

(2) Interval card - This card (see Fig. 54) contains the data which define the vehicle's performance for the upcoming interval. There are 10 variables which

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FIG. 54

PROGRAM INPUT CARDS

Vehicle Card

Column	1	6	11	16	21	26	31	36	41	46	51	56	61
Fortran Symbol	W	AREA	CD	INERTIA	RW	AXLE	PCENT	HPACC	HPMAX	VMAX	ENTYPE	TRTYPE	EFAXLE
Vehicle Parameter	Total Weight (lb)	Frontal Area (ft^2)	Drag Coefficient	Wheel Inertia (slug-ft^2)	Wheel Radius (ft)	Rear Axle Ratio	% Weight on Rear Wheels (percent)	Accessory Horsepower	Maximum Engine Horsepower	Maximum Speed (mph)	Engine Type (see Table IV)	Transmission Type (see Table IV)	Rear Axle Efficiency (decimal)

Interval Card

Column	1	6	11	16	21	26	31	36	41	46
Fortran Symbol	CYCLE (1)	CYCLE (2)	CYCLE (3)	GRADE	MU	F1	F2	F3	DT	DP
Interval Parameter	See Table XXVII			Road Grade (percent)	Friction Coefficient*	Rolling Res. Coefficient (lb/lb)	Rolling Res. Coefficient (lb/lb-fps) x 10^5	Rolling Res. Coefficient (lb/lb-fps^2) x 10^7	Computing Interval Step Size (sec)	Printing Interval Step Size (sec)

Control Card

Column	5	10	15	20	25
Fortran Symbol	NRUN	MODE	NREG	NUAR	NTYP

appear on this card, the first three of which [CYCLE (1), I = 1,3] are described under the section "Control Card." The interval card also contains the road grade (percent), the road-tire adhesion coefficient, and the three rolling resistance coefficients. Since the rolling resistance coefficients are input in units of lb and fps, the coefficients of the terms linear and quadratic in speed are normally quite small and are input times factors of 10^5 and 10^7, respectively. For example, if the coefficient F2 is 2.15×10^{-5} lb/lb-fps, it would be input as just 2.15. Conversion is performed automatically within the program. The computing step size and the printing interval step size are also read in on the interval card, both having units of seconds. The format for the interval card is 16F5.1.

(3) Control card - This card tells the computer which of the nine performance modes the vehicle is to operate in, whether to read additional input cards if either the Federal Driving Cycle or the UARL driving cycles are being used, and whether to read more cards or stop after the current performance interval. The function of each of the five control variables is described below:

Control Variables (NRUN, MODE, NREG, NUAR, AND NTYP)

NRUN - Determines control of program after the present run (the one being read in) is complete. NRUN may ahve only the values 1, 2, or 3, and the functions of each value are given in the table below.

Value of NRUN	Function
1	On next run read new vehicle data, new control data, and new interval data
2	On next run vehicle is the same, just read new control data and new interval data
3	No more runs, stop after this one (the one being read in)

MODE - Determines vehicle performance mode. The first five performance modes (MODE = 1,2,3,4,5) command constant acceleration, and performance is specified by the values assigned to the three subscripted input variables CYCLE(I), I = 1,2,3.

CYCLE (1) always specifies the vehicle's initial speed, in mph, CYCLE (2) specifies either an acceleration or a final speed, and CYCLE (3) is either a final time, a final distance, or an acceleration. These combinations are summarized in Table XXVII. Thus, for example, the user could specify that the vehicle accelerate from 30 mph at an acceleration of 2.5 mph/sec for 3.0 sec by choosing Mode 3 (MODE = 3) and assigning to CYCLE (1), CYCLE (2), and CYCLE (3) the values of 30.0, 2.5, and 3.0, respectively. For each of the first five modes, the computer solves for an equivalent acceleration (a) and final time (t_f), and thus the input values of a and t_f must not be zero. For constant-speed runs, use Mode 1 with $v_i = v_f$ and $t_f > 0$.

Mode 6 (see Table XXVII) commands constant-horsepower acceleration for a given time interval. For this mode, CYCLE (1) is the vehicle's initial speed (mph), CYCLE (2) is the commanded horsepower value, which must be less than or equal to the value of HPMAX on the vehicle card, and CYCLE (3) is the length of time the acceleration is to last.

Modes 7, 8, and 9 command performance over the Federal Register, UARL heavy-duty, and UARL typical cycles, respectively (see Figs. 23, 24, and 25). For these three modes, the computer ignores the values which are input for CYCLE (1), and they may be left blank if desired.

NREG, NUAR, and NTYP - These variables determine whether the data cards on which the driving cycles for Modes 7, 8, and 9 (respectively) are to be read. Input values of NREG, NUAR, and NTYP are given below.

Value of NREG, NUAR or NTYP	Function
0	Read cards for appropriate driving cycle.
≠ 0	Cards for this driving cycle have already been read in.

The values of the control variables for the modes which are not being used (say NREG and NUAR when using Mode 9) are ignored by the computer. For Modes 1-6 the values of NREG, NUAR, and NTYP are all ignored.

TABLE XXVII

PERFORMANCE MODE OPTIONS

Symbol	Meaning
a	Acceleration
t	Time
v	Speed
x	Distance
Subscript	
i	Initial
f	Final

Mode	Description	CYCLE (1)	CYCLE (2)	CYCLE (3)
1	Constant Acceleration	v_i (mph)	v_f (mph)	t_f (sec)
2		v_i (mph)	v_f (mph)	x_f (mi)
3		v_i (mph)	a (mph/sec)	t_f (sec)
4		v_i (mph)	a (mph/sec)	x_f (mi)
5		v_i (mph)	v_f (mph)	a (mph/sec)
6	Constant Horsepower	v_i (mph)	Horsepower	t_f (sec)
7	Federal Driving Cycle	These parameters are ignored for Modes 7 and 8		
8	UARL Heavy-Duty Driving Cycle			
9	UARL Typical Country Driving Cycle			

NOTE: The values a and t_f must not be zero. For constant-speed performance use Mode 1 with $v_i = v_f$ and $t_f > 0$.

The input format for the control card is 5I5. The input variables on the control card have no decimal points and must be right-adjusted; that is, they must appear in the rightmost columns of the spaces to which they have been assigned. For example, if MODE, which occupies columns 6-10 on the control card, has a value of 6, the 6 must appear in column 10.

Other Input Data

In addition to the individual groups of data cards for each performance interval, four other semi-permanent packages of data are required. Three contain the data for the Federal Register and UARL driving cycles and the other contains engine and transmission data. The driving cycle data packages, which comprise 86 cards for the Federal Driving cycle (Mode 7), 176 cards for the UARL heavy-duty cycle (Mode 8), and 145 cards for the UARL typical suburban driving cycle (Mode 9) are input only once for each run. For example, if Mode 7 is to be repeated a number of times for some reason, the data package for Mode 7 is read in only once. If Modes 7, 8, and 9 are not used, the driving cycle data packages are omitted for the run.

Engine and transmission data are brought into the program through a COMMON statement and a BLOCK DATA subprogram. Five sets of data are input in this fashion; the sfc engine data, emissions index values for CO, UHC, and NO_x, and the transmission efficiency data. Each set of data is divided into groups, each containing the data, in x-y pairs, for a particular engine or transmission. The data format for each group is such that it is compatible with the program's independent interpolation subroutine. Basically, the format specifies that the group data shall consist of, in this order: (1) 4 parameters comprising a data ID number, the order of the interpolation (1st, 2nd, or 3rd order), the number of x-values of the data, and the number of y-values of the data (if bivariate), (2) the x-values of the data in ascending order, (3) the y-values of the data (if bivariate) in ascending order, (4) the z-values of the data. The data for each group of a set follow the last member of the preceding group, with the location of the beginning of each group provided to the computer for access to the data.

Use of Input Data Cards

The data cards comprising an input group must be in the following order: vehicle card (if any), control card, and interval card. If Modes 7 and/or 8 or 9 are being run, the appropriate driving cycle data card packages are placed after the interval card for that performance interval. A driving cycle data package is read in only if the driving cycle is being used at least once in the run, and in that case the package is read in only once. All possible groups of input cards are listed below with descriptions of their uses.

Groups of Input Cards

Group	Comments
1. Vehicle card Control card Interval card Duty cycle card	Used only the first time Modes 7 and/or 8 or 9 are being run. Defines a new vehicle. May be used as the first group of input.
2. Control card Interval card Duty cycle cards	Used only the first time Modes 7 and/or 8 or 9 are being run. Uses vehicle data from the preceding interval. Cannot be used for the first group of input.
3. Vehicle card Control card Interval card	Used for Modes 1-6 and for Modes 7 and/or 8 or 9 if they have been run previously. Defines a new vehicle. May be used as the first group of input.
4. Control card Interval card	Used for Modes 1-6 and for Modes 7 and/or 8 or 9 if they have been run previously. Uses vehicle data from the preceding interval. Cannot be used for the first group of input.

The groups of input data may be arranged in any consistent order, and as many groups may be input as desired. Care must be taken, however, that the first input group have a vehicle card as its first member, and that the driving cycle data packages are used for the first, and only the first, time a driving cycle is run.

A complete listing of the automobile performance computer program is given in Figs. 55 and 56.

FIG. 55

COMPUTER PROGRAM LISTING

```
C
C         UARL AUTOMOBILE PERFORMANCE COMPUTER PROGRAM
C
      COMMON/WXYZ/EFAHCD(200),BSFCHP(200),COHP(200),UHCHP(200),
     1   NOXHP(200)
      DIMENSION AREG(1371),AUAR(1081),GUAR(1081),CYCLE(3),NLOOK(4),
     1   ATYP(1155)
      REAL MPHFPS,MPHSEC,MILE,INERTA,MU,MP,MPG
     1 ,NOXHP,NOXS,NOXF,NOXEI
      DATA (NLOOK(I),I=1,4)/1,51,101,151/
C
C         FORMAT STATEMENTS
C
800   FORMAT(16F5.1)
802   FORMAT(5I5)
804   FORMAT(10F8.5)
806   FORMAT(16F5.2)
808   FORMAT(8F10.5)
900   FORMAT(1H1,/11X,27H*** VEHICLE PERFORMANCE ***//)
902   FORMAT(10X,43H* ENGINE 1 (BASIC REGENERATED,FREE TURBINE))
904   FORMAT(10X,51H* ENGINE 2 (SIMPLE CYCLE,SINGLE SHAFT/FREE TURBINE))
906   FORMAT(10X,46H* ENGINE 3 (ADVANCED REGENERATED,SINGLE SHAFT))
908   FORMAT(10X,23H* ENGINE 4 (OTTO CYCLE))
910   FORMAT(10X,36H* TRANSMISSION A (4-SPEED AUTOMATIC))
912   FORMAT(10X,36H* TRANSMISSION B (8-SPEED AUTOMATIC))
914   FORMAT(10X,36H* TRANSMISSION C (3-SPEED AUTOMATIC))
916   FORMAT(10X,44H* TRANSMISSION D (HYDROMECHANICAL AUTOMATIC))
918   FORMAT(16X,18HVEHICLE PARAMETERS//)
920   FORMAT(10X,11HWEIGHT(LB)=,12X,F6.0)
922   FORMAT(10X,21HEQ. WHEEL WEIGHT(LB)=,2X,F6.0)
924   FORMAT(10X,17HDRAG COEFFICIENT=,7X,F5.3)
926   FORMAT(10X,20HFRONTAL AREA(FT**2)=,4X,F5.1)
928   FORMAT(10X,17HWHEEL RADIUS(FT)=,7X,F5.3)
930   FORMAT(10X,11HAXLE RATIO=,13X,F5.3)
932   FORMAT(10X,24HP-CNT WT ON REAR WHEELS=,F5.1)
934   FORMAT(10X,21HACCESSORY HORSEPOWER=,4X,F4.1)
936   FORMAT(10X,17HRATED HORSEPOWER=7X,F5.1//)
938   FORMAT(//10X,19HINTERVAL PARAMETERS,9H ** MODE ,I1//)
940   FORMAT(10X,15HPER CENT GRADE=,8X,F6.1)
942   FORMAT(10X,21HFRICTION COEFFICIENT=,4X,F4.2)
944   FORMAT(10X,20HACCEL REQD(MPH/SEC)=,4X,F5.2//)
946   FORMAT(10X,18HENGINE HORSEPOWER=,6X,F5.1//)
948   FORMAT(37X,3HACC,4X,4HAERO,3X,4HROLL,7H  GRADE,3X,4HTRAC,2X,
     1 5HTOTAL,4X,4HFUEL,2X,24H * * * POLLUTANTS * * * )
950   FORMAT(3X,4HTIME,3X,3HACC,5X,5HSPEED,3X,4HDIST,3X,3HRHO,
     1 6(2X,5HPOWER),4X,4HFLOW,5X,2HCO,6X,3HUHC,5X,3HNOX)
952   FORMAT(7H  (SEC),16H (MPH/S) (MPH),3X,4H(MI),6X,
     1 6(3X,4H(HP)),8H (LB/HR),8X,12H(GRAMS/HOUR))
954   FORMAT(1X,F6.1,F7.2,F7.1,F7.3,F6.2,6F7.1,F8.1,2X,3F8.3)
956   FORMAT(//18X,35H***** FUEL AND EMISSIONS DATA *****//)
958   FORMAT(
     1 10X,17HFUEL USED (LB) = ,F7.3,5X,16HMILEAGE (MPG) = ,F7.3/
     2 10X,17HCO (GM)        = ,F7.3,5X,16HCO (GPM)      = ,F7.3/
     3 10X,17HUHC (GM)       = ,F7.3,5X,16HUHC (GPM)     = ,F7.3/
     4 10X,17HNOX (GM)       = ,F7.3,5X,16HNOX (GPM)     = ,F7.3/
     5 10X,17HMOTUS PER MILE = ,F7.0//
     6 10X,17HAVE HORSEPOWER = ,F7.3)
C
C     LIST CONSTANTS
C
      G=32.2
      SMALL=1.0E-08
      MPHFPS=1.46666
      FPSMPH=0.681818
      MPHSEC=0.04555
      RPMRPS=0.1047198
      RPSRPM=9.549297
      RADDEG=57.29578
      MILE=5280.
      DENS=2.3723E-03
C
206   CONTINUE
C
C         READ VEHICLE DATA
C
      READ(5,800)W,HPACC,HPMAX,ENTYPE,TRTYPE
      AREA=25.0
      CD=0.48
      INERTA=0.0
      RW=1.1
      AXLE=3.0
      PCENT=50.
      VMAX=100.
      EFAXLE=0.95
      NTYPE=ENTYPE+0.1
      NTRANS=TRTYPE+0.1
C
C         COMPUTE VEHICLE PARAMETERS
C
      WW=4.0*INERTA*G/(RW**2)
      WP=W+WW
      FD=CD*AREA*DENS
      MP=WP/G
C
C         INITIALIZE RUN PARAMETERS AND PRINT VEHICLE PARAMETERS
C
      TS=0.0
      X=0.0
      TSL=0.0
      XL=0.0
      WRITE(6,900)
      WRITE(6,918)
      GO TO(701,702,703,704),NTYPE
701   WRITE(6,902)
      GO TO 207
702   WRITE(6,904)
      GO TO 207
703   WRITE(6,906)
      GO TO 207
704   WRITE(6,908)
207   CONTINUE
      GO TO (711,712,713,714),NTRANS
711   WRITE(6,910)
      GO TO 208
712   WRITE(6,912)
      GO TO 208
713   WRITE(6,914)
      GO TO 208
714   WRITE(6,916)
      GO TO 208
208   CONTINUE
      WRITE(6,920)W
      WRITE(6,922)WW
      WRITE(6,924)CD
      WRITE(6,926)AREA
      WRITE(6,928)RW
      WRITE(6,930)AXLE
      WRITE(6,932)PCENT
      WRITE(6,934)HPACC
      WRITE(6,936)HPMAX
209   CONTINUE
C
C         READ CONTROL AND INTERVAL DATA
C
      READ(5,802)NRUN,MODE,NREG,NUAR,NTYP
      READ(5,800) (CYCLE(I),I=1,3),GRADE,DT,DP
      MU=1.0
      F1=0.015
      F2=2.15E-05
      F3=1.85E-07
C
C         COMPUTE MODE INDEPENDENT PARAMETERS
C
      FGRAV=W*GRADE/100.
      ALPHAR=ASIN(GRADE/100.)
      ALPHAD=ALPHAR*RADDEG
      WN=W*COS(ALPHAR)
      FMAX=MU*WN*PCENT/100.
      FR1=F1*WN
      FR2=F2*WN
      FR3=F3*WN
      NPRINT=DP/DT+.001
C
C         COMPUTE MODE DEPENDENT PARAMETERS
C
      GO TO (210,211,212,213,214,215,216,217,218),MODE
210   CONTINUE
C
C         MODE 1
C
      VI1=CYCLE(1)
      VF1=CYCLE(2)
      TAUF=CYCLE(3)
      VI2=VI1*MPHFPS
      VF2=VF1*MPHFPS
      AR2=(VF2-VI2)/TAUF
      AR1=AR2*FPSMPH
      GO TO 219
211   CONTINUE
C
C         MODE 2
C
      VI1=CYCLE(1)
      VF1=CYCLE(2)
      XF=CYCLE(3)
      VI2=VI1*MPHFPS
      VF2=VF1*MPHFPS
      AR2=(VF2**2-VI2**2)/(2.*XF*MILE)
      AR1=AR2*FPSMPH
      TAUF=(VF2-VI2)/AR2
      GO TO 219
212   CONTINUE
C
C         MODE 3
C
      VI1=CYCLE(1)
      AR1=CYCLE(2)
      TAUF=CYCLE(3)
      AR2=AR1*MPHFPS
      VI2=VI1*MPHFPS
      VF2=VF1*MPHFPS
      GO TO 219
213   CONTINUE
C
C         MODE 4
C
      VI1=CYCLE(1)
      AR1=CYCLE(2)
      XF=CYCLE(3)
      AR2=AR1*MPHFPS
      VI2=VI1*MPHFPS
      VF2=VI2**2+2.*AR2*XF*MILE
      VF2=SQRT(VF2)
      TAUF=(VF2-VI2)/AR2
      GO TO 219
214   CONTINUE
C
C         MODE 5
C
      VI1=CYCLE(1)
      VF1=CYCLE(2)
      AR1=CYCLE(3)
      VI2=VI1*MPHFPS
      VF2=VF1*MPHFPS
      AR2=AR1*MPHFPS
      TAUF=(VF1-VI1)/AR1
      GO TO 219
215   CONTINUE
C
C         MODE 6
C
      VI1=CYCLE(1)
      HPCON=CYCLE(2)
      TAUF=CYCLE(3)
      VI2=VI1*MPHFPS
      GO TO 219
216   CONTINUE
C
C         MODE 7
C
      VI1=0.0
      VI2=0.0
      TAUF=1370.
      IF(NREG.GT.0)GO TO 219
      READ(5,800) (AREG(I),I=1,1371)
      GO TO 219
217   CONTINUE
C
C         MODE 8
C
```

COMPUTER PROGRAM LISTING (CONT'D)

```
      VI1=0.0
      VI2=0.0
      TAUF=1080.
      IF(NUAR.GT.0)GO TO 219
      READ(5,804) (AUAR(I),I=1,1081)
      READ(5,806) (GUAR(I),I=1,1081)
      GO TO 219
218   CONTINUE
C
C        MODE 9
C
      VI1=0.0
      VI2=0.0
      TAUF=1154.
      IF(NTYP.GT.0)GO TO 219
      READ(5,808) (ATYP(I),I=1,1155)
219   CONTINUE
C
C        INITIALIZE AND WRITE HEADING
C
      IJK=1
      FUELF=0.0
      FUELS=0.0
      COF=0.0
      CCOS=0.0
      UHCF=0.0
      UHCS=0.0
      NOXF=0.0
      NOXS=0.0
      HPAREA=0.0
      HPSUM=0.0
      DELTA=DT
      TAU=-DELTA
      V1=VI1
      V2=VI2
      XI=0.0
      KPRINT=NPRINT
      NEND=1
      WRITE(6,938)MODE
      WRITE(6,940)GRADE
      WRITE(6,942)MU
      IF(MODE.EQ.6)GO TO 220
      WRITE(6,944)AR1
      GO TO 221
220   CONTINUE
      WRITE(6,946)HPCON
221   CONTINUE
      WRITE(6,948)
      WRITE(6,950)
      WRITE(6,952)
222   CONTINUE
C
C        COMPUTE PRESENT TIME,SPEED AND DISTANCE
C
      TAU=TAU+DELTA
      IF(TAU.LT.SMALL)GO TO 223
      KPRINT=KPRINT+1
      VL1=V1
      VL2=V2
      V1=VL1+A1*DELTA
      V2=VL2+A2*DELTA
      XI=XI+(VL2*DELTA+0.5*A2*DELTA**2)/MILE
223   CONTINUE
      FAERO=0.5*FD*V2**2
      FROLL=FR1+FR2*V2+FR3*V2**2
      IF(V2.LT.SMALL)FROLL=0.0
      IF(MODE.EQ.6)GO TO 229
C
C        COMPUTATIONS FOR CONSTANT ACCELERATION
C
      GO TO (228,228,228,228,228,229,224,225,226),MODE
224   CONTINUE
      AR1=AREG(IJK)
      AR2=AR1*MPHFPS
      GO TO 228
225   CONTINUE
      AR1=AUAR(IJK)
      GRADE=GUAR(IJK)
      AR2=AR1*MPHFPS
      FGRAV=.01*W*GRADE
      ALPHAR=ASIN(.01*GRADE)
      WN=W*COS(ALPHAR)
      FMAX=.01*MU*WN*PCENT
      FR1=F1*WN
      FR2=F2*WN
      FR3=F3*WN
      FROLL=FR1+FR2*V2+FR3*V2**2
      IF(V2.LT.SMALL)FROLL=0.0
      GO TO 228
226   CONTINUE
      AR1=ATYP(IJK)
      AR2=AR1*MPHFPS
228   CONTINUE
      A1=AR1
      A2=AR2
      FINER=MP*A2
      FPROP=FINER+FAERO+FROLL+FGRAV
      RHO=FPROP/FMAX
      IF(RHO.LE.1.0)GO TO 312
C
C        TRACTION LIMITED COMPUTATIONS (CONSTANT ACCELERATION)
C
      FPROP=FMAX
      RHO=1.0
      FINER=FPROP-FAERO-FROLL-FGRAV
      A2=FINER/MP
      A1=A2*FPSMPH
312   TW=FPROP*RW
      TSH=TW/AXLE
      RPSW=V2/RW
      RPMW=RPSW*RPSRPM
      RPMSH=RPMW*AXLE
      HPSH=FPROP*V2/550.
      VFRAC=V1/VMAX
      CALL LOOKUP(EFABCD,NLOOK(NTRANS),VFRAC,0.0,EFF)
      HPENG=HPSH/(EFF*EFAXLE)+HPACC
      IF(HPENG.LT.HPMAX) GO TO 231
      HPENG=HPMAX
      HPSH=(HPENG-HPACC)*EFF*EFAXLE
      FPROP=HPSH*550./V2
      RHO=FPROP/FMAX
      FINER=FPROP-FAERO-FROLL-FGRAV
      A2=FINER/MP
      A1=A2*FPSMPH
      TW=FPROP*RW
      TSH=TW/AXLE
      GO TO 231
229   CONTINUE
```

```
C
C        COMPUTATIONS FOR CONSTANT HORSEPOWER
C
      RPSW=V2/RW
      IF(V2.LT.SMALL)V2=SMALL
      RPMW=RPSW*RPSRPM
      RPMSH=RPMW*AXLE
      VFRAC=V1/VMAX
      CALL LOOKUP(EFABCD,NLOOK(NTRANS),VFRAC,0.0,EFF)
      HPSH=(HPCON-HPACC)*EFF*EFAXLE
      FPROP=HPSH*550./V2
      RHO=FPROP/FMAX
      IF(RHO.GT.1.0)GO TO 230
      FINER=FPROP-FAERO-FROLL-FGRAV
      A2=FINER/MP
      A1=A2*FPSMPH
      HPENG=HPCON
      GO TO 311
230   CONTINUE
C
C        TRACTION LIMITED COMPUTATIONS (CONSTANT HORSEPOWER)
C
      FPROP=FMAX
      RHO=1.0
      HPSH=FPROP*V2/550.
      HPENG=HPSH/(EFF*EFAXLE)+HPACC
      FINER=FPROP-FAERO-FROLL-FGRAV
      A2=FINER/MP
      A1=A2*FPSMPH
311   TW=FPROP*RW
      TSH=TW/AXLE
231   CONTINUE
      IF(HPENG.LT.HPACC)HPENG=HPACC
      QQ=DELTA/3600.
      FUELS=FUELS+FUELF*QQ
      CCOS=CCOS+COF*QQ
      UHCS=UHCS+UHCF*QQ
      NOXS=NOXS+NOXF*QQ
      IJK=IJK+1
      HPVAL=HPENG/HPMAX
C
C        LOOK UP SFC AND EMISSIONS INDICES FROM TABULAR DATA.
C        USE LINEARIZED FORMULAS FOR FUEL FLOW IF NORMALIZED
C        HORSEPOWER IS LESS THAN SPECIFIED VALUES.
C
      GO TO (232,234,236,238),NTYPE
232   CONTINUE
      IF(HPVAL.LT.0.0666667)GO TO 233
      GO TO 240
233   CONTINUE
      FUELF=10.5*(HPMAX/150.)+0.368*HPENG
      GO TO 241
234   CONTINUE
      IF(HPVAL.LT.0.0666667)GO TO 235
      GO TO 240
235   CONTINUE
      FUELF=12.75*(HPMAX/150.)+0.545*HPENG
      GO TO 241
236   CONTINUE
      IF(HPVAL.LT.0.0666667)GO TO 237
      GO TO 240
237   CONTINUE
      FUELF=8.4*(HPMAX/150.)+0.315*HPENG
      GO TO 241
238   CONTINUE
      IF(HPVAL.LT.0.0666667)GO TO 239
      GO TO 240
239   CONTINUE
      FUELF=6.0*(HPMAX/150.)+0.6*HPENG
      GO TO 241
240   CONTINUE
      CALL LOOKUP(BSFCHP,NLOOK(NTYPE),HPVAL,0.0,BSFC)
      FUELF=BSFC*HPENG
241   CONTINUE
      CALL LOOKUP(COHP  ,NLOOK(NTYPE),HPVAL,0.0,COEI)
      CALL LOOKUP(UHCHP ,NLOOK(NTYPE),HPVAL,0.0,UHCEI)
      CALL LOOKUP(NOXHP ,NLOOK(NTYPE),HPVAL,0.0,NOXEI)
C
C        COMPUTE EMISSIONS DATA
C
      COF=COEI*FUELF*0.4536
      UHCF=UHCEI*FUELF*0.4536
      NOXF=NOXEI*FUELF*0.4536
      IF(NEND.EQ.2)GO TO 245
242   CONTINUE
      IF(KPRINT.EQ.NPRINT)GO TO 245
243   CONTINUE
      TEST=TAUF-TAU
      IF(TEST.GT.DT)GO TO 244
      DELTA=TEST
      NEND=2
244   CONTINUE
      HPAREA=HPENG*DELTA
      HPSUM=HPSUM+HPAREA
      GO TO 222
245   CONTINUE
C
C        COMPUTE OUTPUT AND PRINT
C
      KPRINT=0
      TS=TSL+TAU
      X=XL+XI
      HPINER=FINER*V2/550.
      HPAERO=FAERO*V2/550.
      HPROLL=FROLL*V2/550.
      HPGRAV=FGRAV*V2/550.
      WRITE(6,954)TAU,A1,V1,XI,RHO,HPINER,HPAERO,HPROLL,HPGRAV,
     1 HPSH,HPENG,FUELF,COF,UHCF,NOXF
      GO TO (243,246),NEND
246   CONTINUE
      GAL=FUELS*0.14286
      BTUS=FUELS*18400.
      GPMCO=CCOS/XI
      GPMUHC=UHCS/XI
      GPMNOX=NOXS/XI
      MPG=XI/GAL
      BTUSMI=BTUS/XI
      HPAV=HPSUM/TAU
      WRITE(6,956)
      WRITE(6,958)FUELS,MPG,CCOS,GPMCO,UHCS,GPMUHC,NOXS,GPMNOX,
     1 BTUSMI,HPAV
      TSL=TS
      XL=X
      GO TO (206,209,247),NRUN
247   CONTINUE
      STOP
      END
```

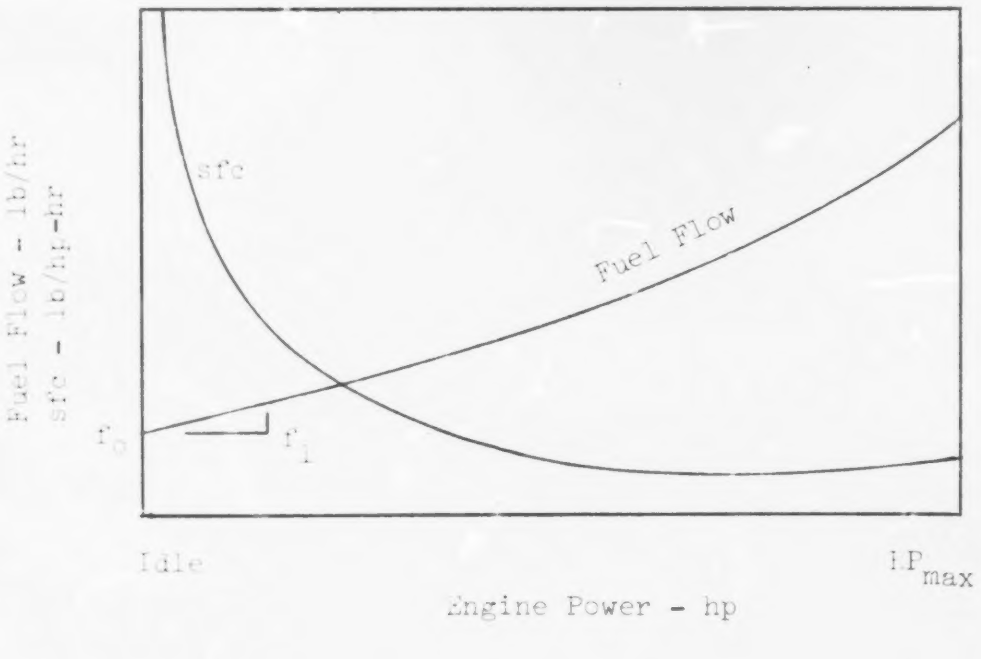
APPENDIX II

FUEL CONSUMPTION DATA FOR LOW OUTPUT POWER

The fuel consumption data input to the automobile performance computer program are in the form of sfc curves versus normalized horsepower (Fig.28) so that data from a base-line engine with a given rated power could be used for other engines with rated power within, say, ± 20% of the base-line engine. As an example of the use of normalized horsepower, consider two similar engines whose rated maximum output powers differ by 20%. At the same fraction of total output power (normalized horsepower), both engines would have the same sfc (lb/hp-hr), but the fuel flow (lb/hr) for the larger engine would be 20% greater, because fuel flow equals sfc times horsepower. For both engines operating at the same power levels, the larger engine would have a greater sfc than the smaller engine, assuming the sfc decreases with increasing power level, because its normalized power level is lower, and hence its fuel flow would be greater.

Below normalized horsepower values of about 0.0333 (5 hp on a 150-hp engine), the sfc curves (Fig.28) become so steep as to be impractical for use as tabular input. Although fuel flow data are not directly usable for engine scaling, they do remain finite as output power goes to zero (idle), and the technique described below was used so that fuel flow data (not sfc) could be employed to determine fuel consumption for engines larger or smaller than the base-line size.

The technique can be illustrated with the sketch below which shows fuel flow and sfc curves for a typical small (150-hp) gas turbine or Otto-cycle engine.



The fuel flow and sfc values are related to each other by Eq. (A-1)

$$\text{Fuel flow} = \text{sfc} \times \text{HP}, \tag{A-1}$$

where HP = output power (hp)

Now in the region of zero output power (idle), the fuel flow curve can be approximated by a linear equation of the form

$$\text{Fuel flow} = f_0 + f_1 (\text{HP}), \tag{A-2}$$

where f_0 = idle fuel flow (lb/hr)

f_1 = slope of fuel flow curve at idle power.

Employing the definition of sfc,

$$\text{sfc} = \frac{f_0}{\text{HP}} + f_1. \tag{A-3}$$

Normalized horsepower, h, for the base-line engine described by the curves above is defined by

$$h = \text{HP}/\text{HP}_{\text{max}_0}\ , \tag{A-4}$$

where HP_{max_0} = maximum power of the base-line engine (hp),

and putting Eq. (A-4) into Eq. (A-3):

$$\text{sfc} = \frac{f_0}{h\ \text{HP}_{\text{max}_0}} + f_1, \tag{A-5}$$

which is true for engines of the base-line engine design with maximum horsepower near HP_{max_0}. For these other engines, normalized hp can be written as

$$h = HP/HP_{max} \tag{A-6}$$

where HP_{max} = maximum engine power (hp)

Putting Eq. (A-6) into Eq. (A-5) yields the general expression

$$sfc = \left(\frac{HP_{max}}{HP_{max_0}}\right)\frac{f_0}{HP} + f_1 \text{ ,} \tag{A-7}$$

so that substituting Eq. (A-7) into Eq. (A-1), for engines close to the base-line engine size,

$$\text{Fuel flow} = \left(\frac{HP_{max}}{HP_{max_0}}\right) f_0 + f_1 HP. \tag{A-8}$$

Thus, for low power levels and engines near the base-line engine size, the slope of the fuel flow curve is the same, but the idle fuel flow varies as the ratio of the maximum engine power to the maximum base-line engine power. Hence, the effect of increasing or decreasing engine size is represented by a vertical translation of the fuel flow curve. The values f_0, f_1, and HP_{max_0} serve to define the fuel consumption characteristics for all (similar) engines near the base-line size when HP_{max} is known. Generally $0.8\ HP_{max_0} < HP_{max} < 1.2\ HP_{max_0}$.

APPENDIX III

DOT HIGH-SPEED PASS MANEUVER

The DOT High-Speed Pass Maneuver, as it appears in Ref. 11, is as follows: The maximum time and maximum distance to go from an initial speed of 50 mph with the front of the vehicle 100 ft behind the back of a 55-ft truck traveling at a constant 50 mph to a position where the back of the vehicle is 100 ft in front of the front of the 55-ft truck is 15 sec and 1400 ft. The entire maneuver takes place in a traffic lane adjacent to the lane in which the truck is operated. The vehicle speed during the maneuver cannot exceed 80 mph. Ambient conditions are 14.7 psia, 85 F. Vehicle weight is W_{test}, and acceleration is on level grade.

Although the DOT pass maneuver is not programmed as a specific performance mode option in the automobile performance computer program, it is possible to determine the required passing time and distance for a given vehicle based on output data from the program's constant-power option. Maximum vehicle acceleration (maximum hp) is commanded and the time and distance to go from 50 to 80 mph is recorded from the printout. These data are then substituted into the simple algebraic equations developed below to obtain the total passing time and distance for the maneuver.

The geometry of the pass maneuver is shown in Fig. 57 (not to scale). All times are in seconds, all distances are in feet, and the length of the vehicle performing the pass maneuver is taken to be 17 ft. It is assumed that the passing vehicle accelerates to 80 mph, and then continues at a constant 80 mph until the maneuver is complete. Let the total time and distance of the pass be denoted by t_{pass} and x_{pass}, respectively, and let t_a and x_a be the time and distance the passing vehicle requires to accelerate from 50 to 80 mph as determined from the computer printout. Further, let x_{car} and x_{truck} be the coordinates of the fronts of the passing vehicle and truck as measured from the front of the passing vehicle at the beginning of the maneuver, and let t represent the elapsed time as measured from the beginning of the maneuver. Noting that 50 mph = 220/3 fps and 80 mph = 352/3 fps, one obtains

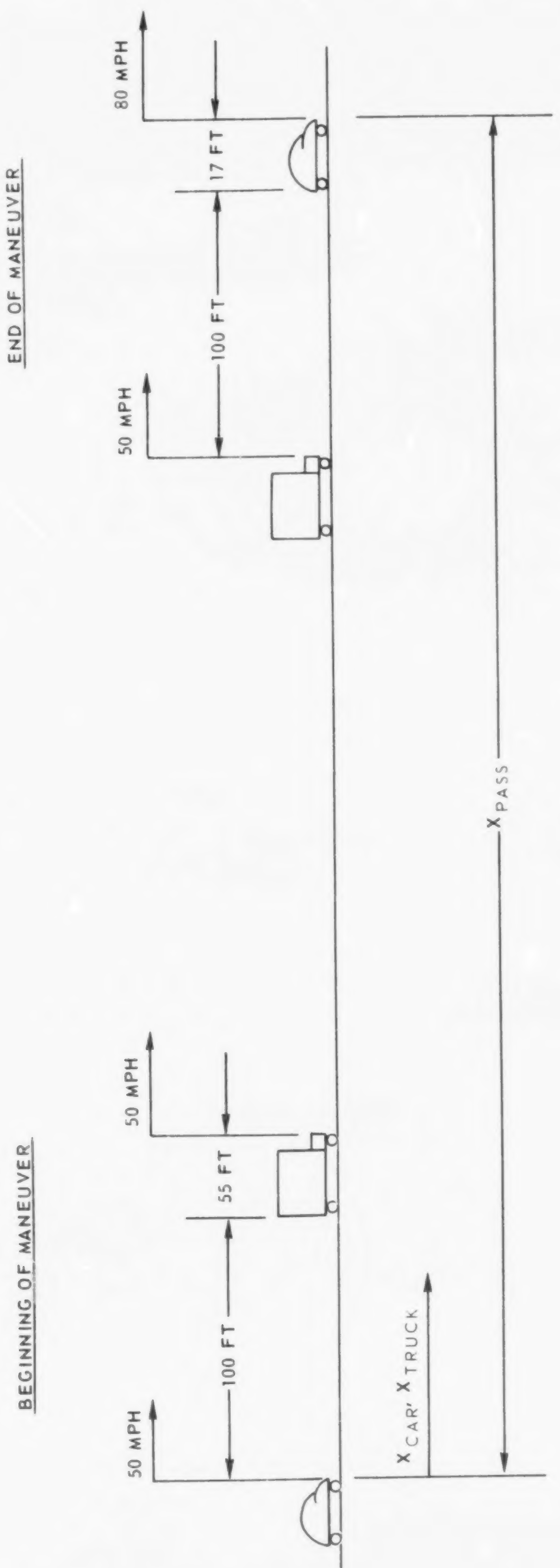
$$x_{truck} = 155 + 220/3\ t \qquad \text{(A-9)}$$

$$x_{car} = x_a + 352/3\ (t-t_a) \text{ for } t \geq t_a \qquad \text{(A-10)}$$

FIG. 57

DOT HIGH-SPEED PASS MANEUVER GEOMETRY

(NOT TO SCALE)


BEGINNING OF MANEUVER
50 MPH
100 FT
55 FT
50 MPH
X_{CAR}, X_{TRUCK}
END OF MANEUVER
50 MPH
100 FT
17 FT
80 MPH
X_{PASS}

From Fig. 57, the maneuver is complete when

$$x_{car} = x_{truck} + 117 \qquad (\text{at } t = t_{pass}) \tag{A-11}$$

Substituting Eqs. (A-9) and (A-10) into (A-11), and solving for t_{pass},

$$t_{pass} = \frac{272 + 352/3\ t_a - x_a}{44}, \tag{A-12}$$

and from Eqs. (A-10) and (A-12),

$$x_{pass} = x_a + 352/3\ (t_{pass} - t_a). \tag{A-13}$$

Equations (A-12) and (A-13) define the total passing time and distance when t_a and x_a are known.

APPENDIX IV

DETAILED COST DRAWINGS

Details of selected components and assemblies suitable for manufacturing cost estimation are reproduced in Figs. 58 through 83. These components were selected jointly by UARL and UACL on the basis that they represented the major portion of the costs associated with the SSS-12 and SFT-12 engines.

These details were then supplied to appropriate potential vendors for cost and price estimation and were also subjected to a production methods analysis at UACL. As a result of vendor suggestions and the contractor's manufacturing experience, certain material specification and process changes were made in order to derive the final probable manufacturing cost estimates. Thus, the material specifications found in the manufacturing cost estimates (Tables X and XI) may not agree in every case with those shown by the drawings.

FIG. 58

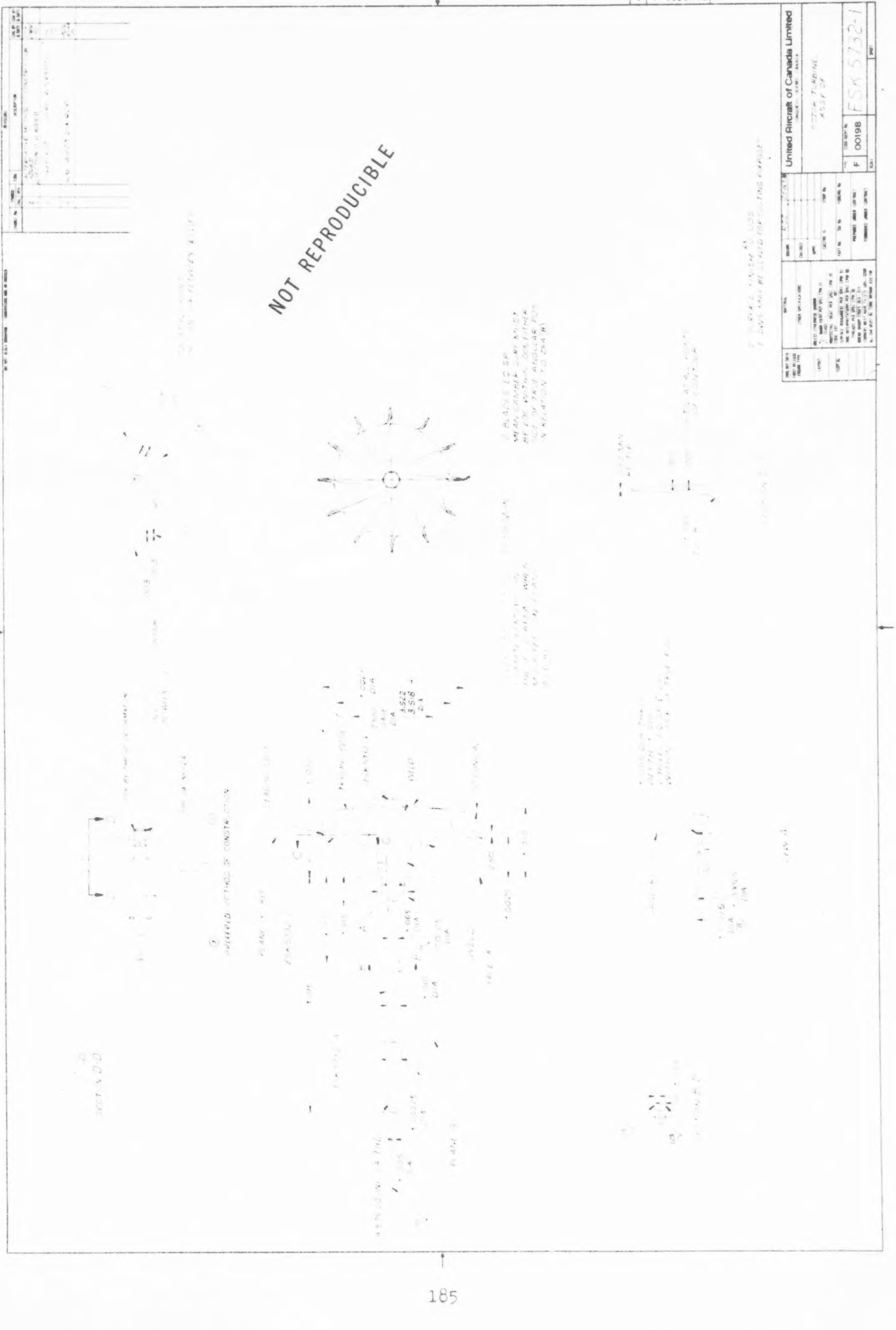

NOT REPRODUCIBLE
ESK5732-1
United Aircraft of Canada Limited
ROTOR TURBINE ASSY OF
F
00198
FSK 5732-1

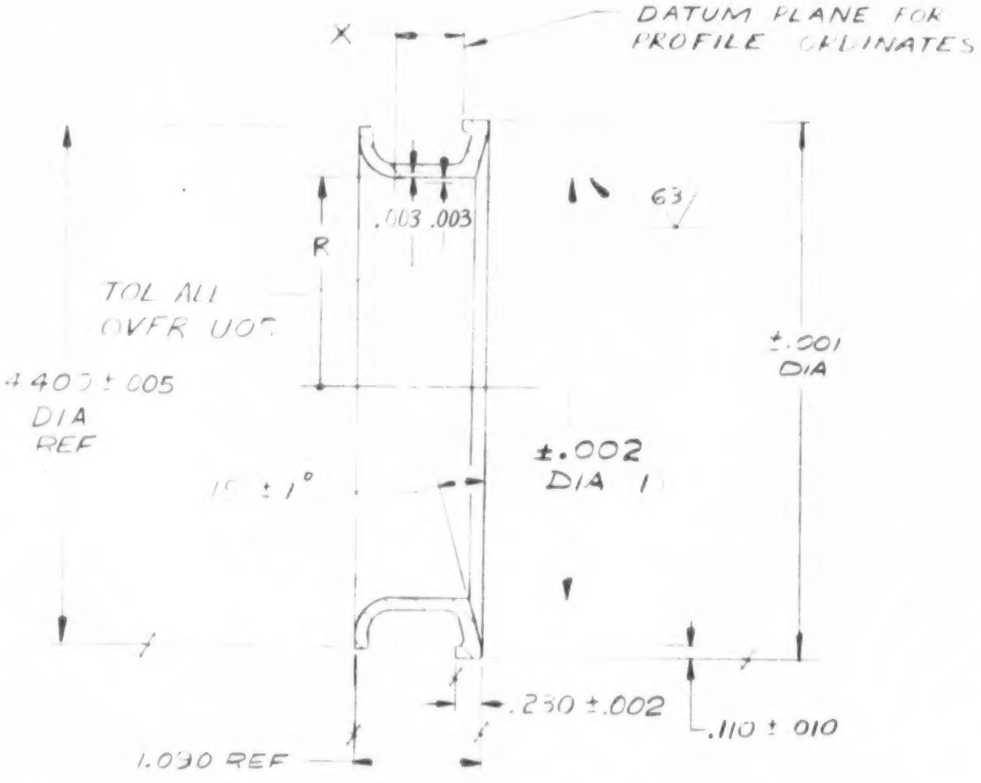
FIG. 59

K-971017-4

ESK 5732-5

DO NOT SCALE DRAWING   DIMENSIONS ARE IN INCHES

REVISIONS					
CHANGE No	SYMBOL		DESCRIPTION	CHG BY & DATE	CHK BY & DATE
	CHG	REV			
			±.002 ADDED	MDS 9 JUN 71	


X
DATUM PLANE FOR
PROFILE ORDINATES
.003 .003
R
63
TOL ALL
OVER UOS
4.400 ±.005
DIA
REF
±.001
DIA
±.002
DIA 1
15 ±1°
.230 ±.002
.110 ±.010
1.030 REF

1—DWG MAY BE SCALED FOR COSTING PURPOSES

DWG REF DATA FIRST RELEASE ENGINE TYPE	MATERIAL	UNLESS OTHERWISE SHOWN MARK IDENT PER SPEC CPW 10 CLASS	DRAWN	RGC	16 OCT 70
LAYOUT	OTHER SPECIFICATIONS	PROTECTIVE TREAT PER SPEC CPW 20 CODE EXT INT	CHECKED		
PART No	CASTING No STRIP No	SURFACE ROUGHNESS PER SPEC CPW 31 DWG INTERPRETATIONS PER SPEC CPW 90 THREADS PER SPEC CPW 30	APPD		
DCB No	PATT No DIE No FORG No	BREAK SHARP EDGES .003—.015 CORNERS MUST HAVE FILLETS .005—.020R ALL DIA MUST BE CONC WITHIN .010 FIR			

United Aircraft of Canada Limited
LONGUEUIL, QUEBEC, CANADA

SHROUD-TURBINE

SIZE	CODE IDENT No	
B	00198	ESK 5732-5
SCALE	WEIGHT LB	SHEET

FIG. 60

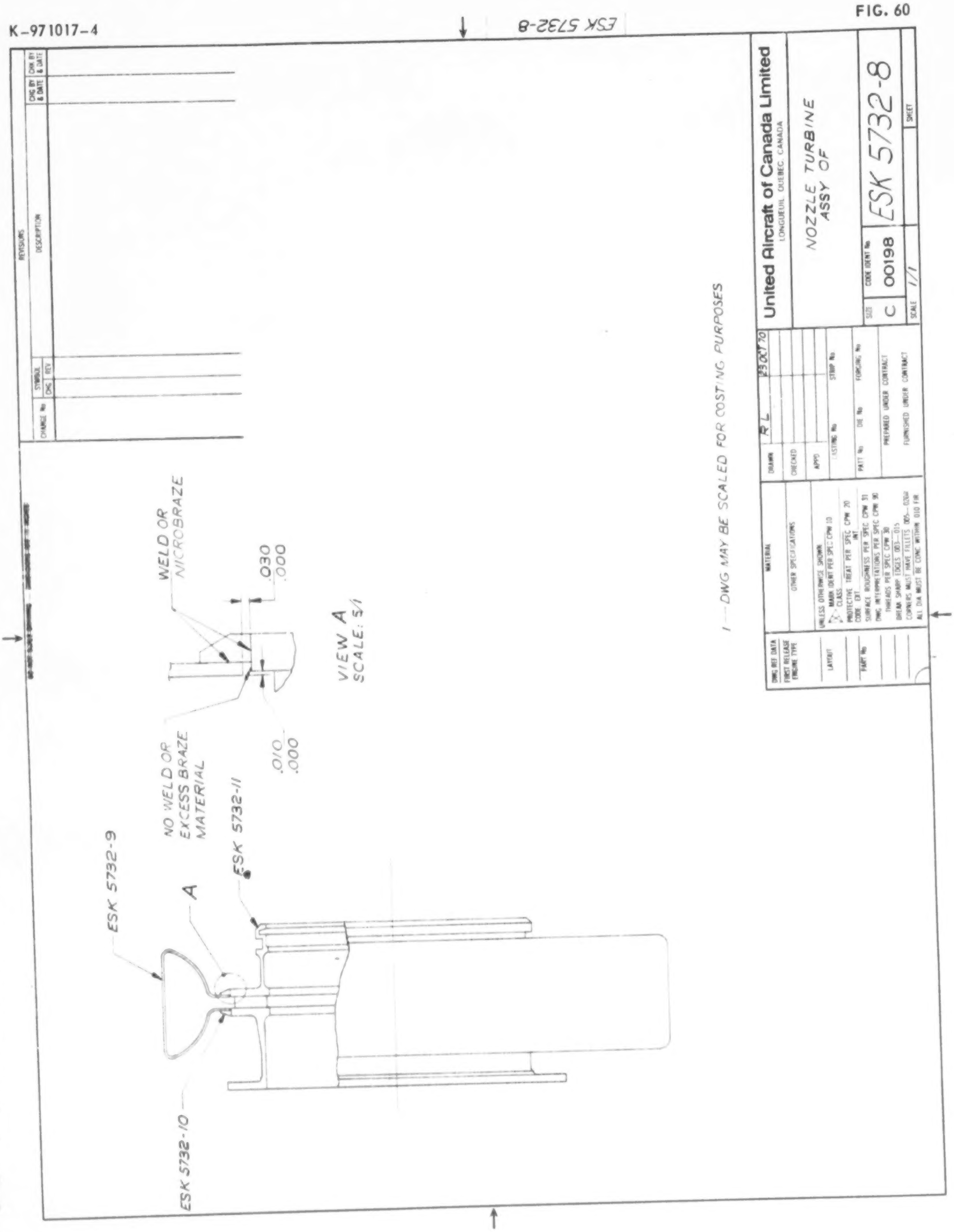

ESK 5732-8
REVISIONS
CHANGE No
SYMBOL
DWG
REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
ESK 5732-9
A
ESK 5732-10
ESK 5732-11
NO WELD OR EXCESS BRAZE MATERIAL
WELD OR NICROBRAZE
.030 .000
.010 .000
VIEW A
SCALE: 5/1
1—DWG MAY BE SCALED FOR COSTING PURPOSES
United Aircraft of Canada Limited
LONGUEUIL, QUEBEC, CANADA
NOZZLE TURBINE ASSY OF
DRAWN R L 23 OCT 70
CHECKED
APPD
CASTING No
STRIP No
PATT No
DIE No
FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
SIZE C
CODE IDENT No 00198
ESK 5732-8
SCALE 1/1
SHEET
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003—.015
CORNERS MUST HAVE FILLETS .005—.020
ALL DIA MUST BE CONC WITHIN .010 FIR
DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No

FIG. 61

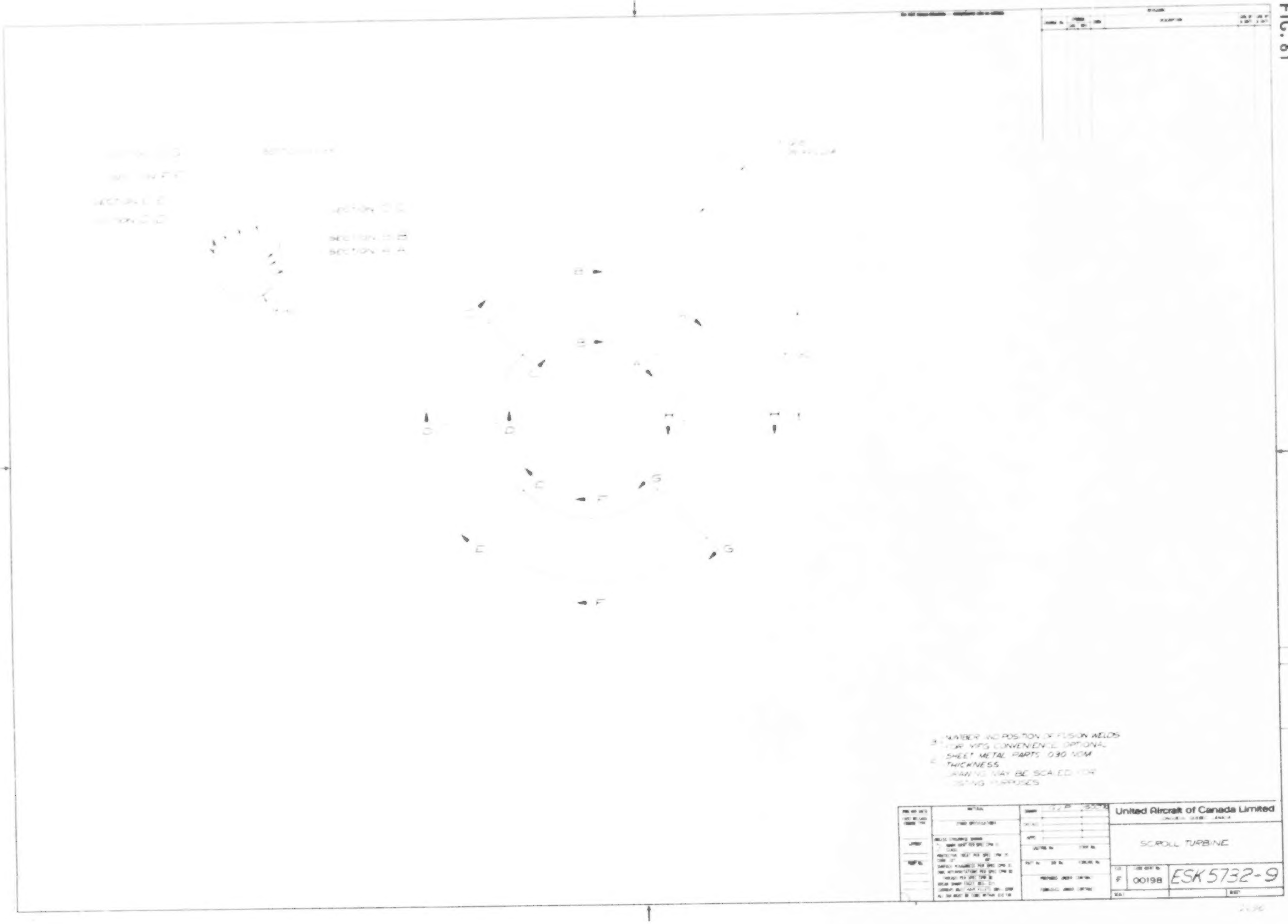

SECTION C-C
SECTION B-B
SECTION A-A
A
B
C
D
E
F
G
H
3. NUMBER AND POSITION OF FUSION WELDS FOR MFG CONVENIENCE OPTIONAL
2. SHEET METAL PARTS .030 NOM THICKNESS
DRAWING MAY BE SCALED FOR COSTING PURPOSES
United Aircraft of Canada Limited
SCROLL TURBINE
F
00198
ESK 5732-9

FIG. 62

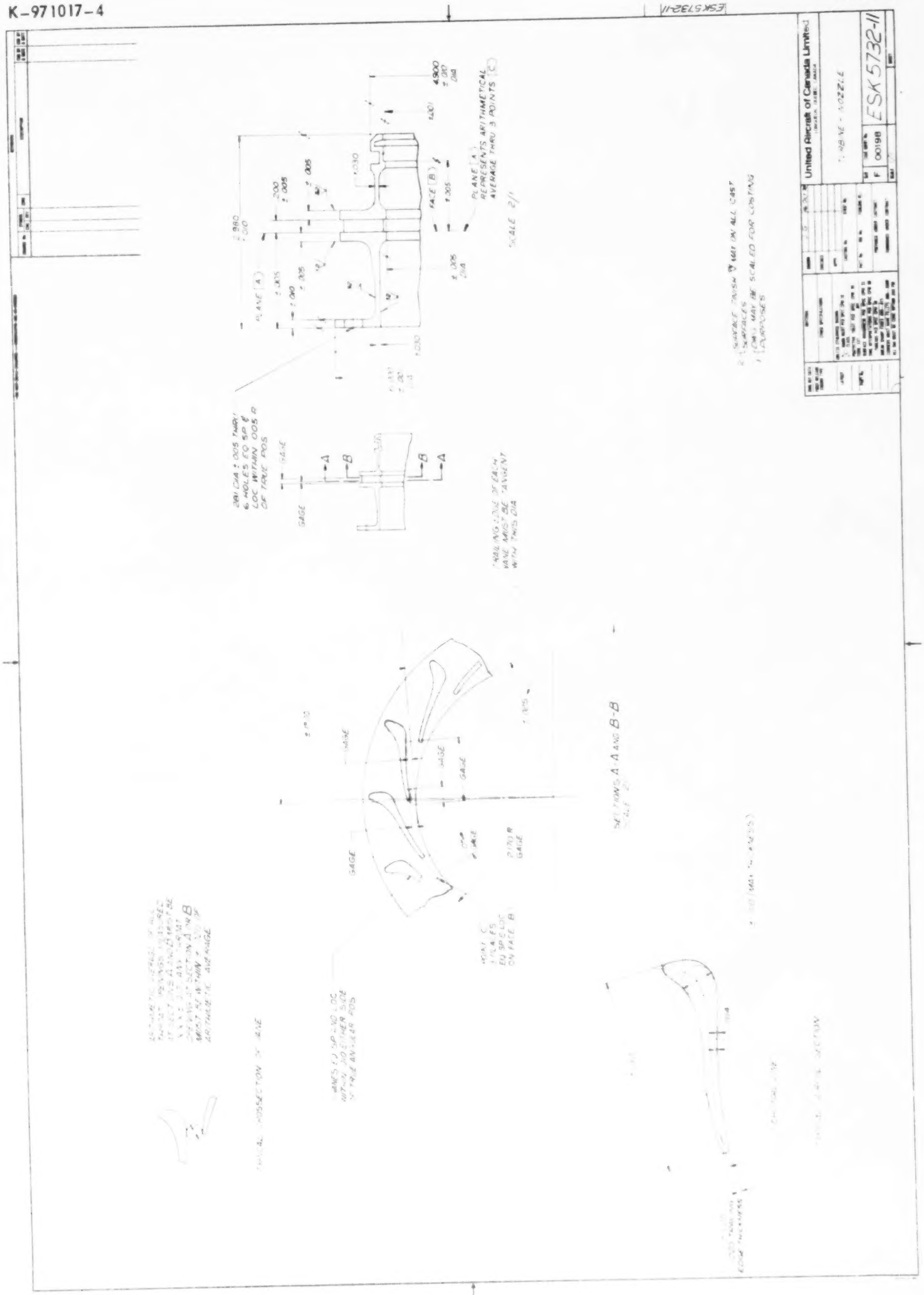
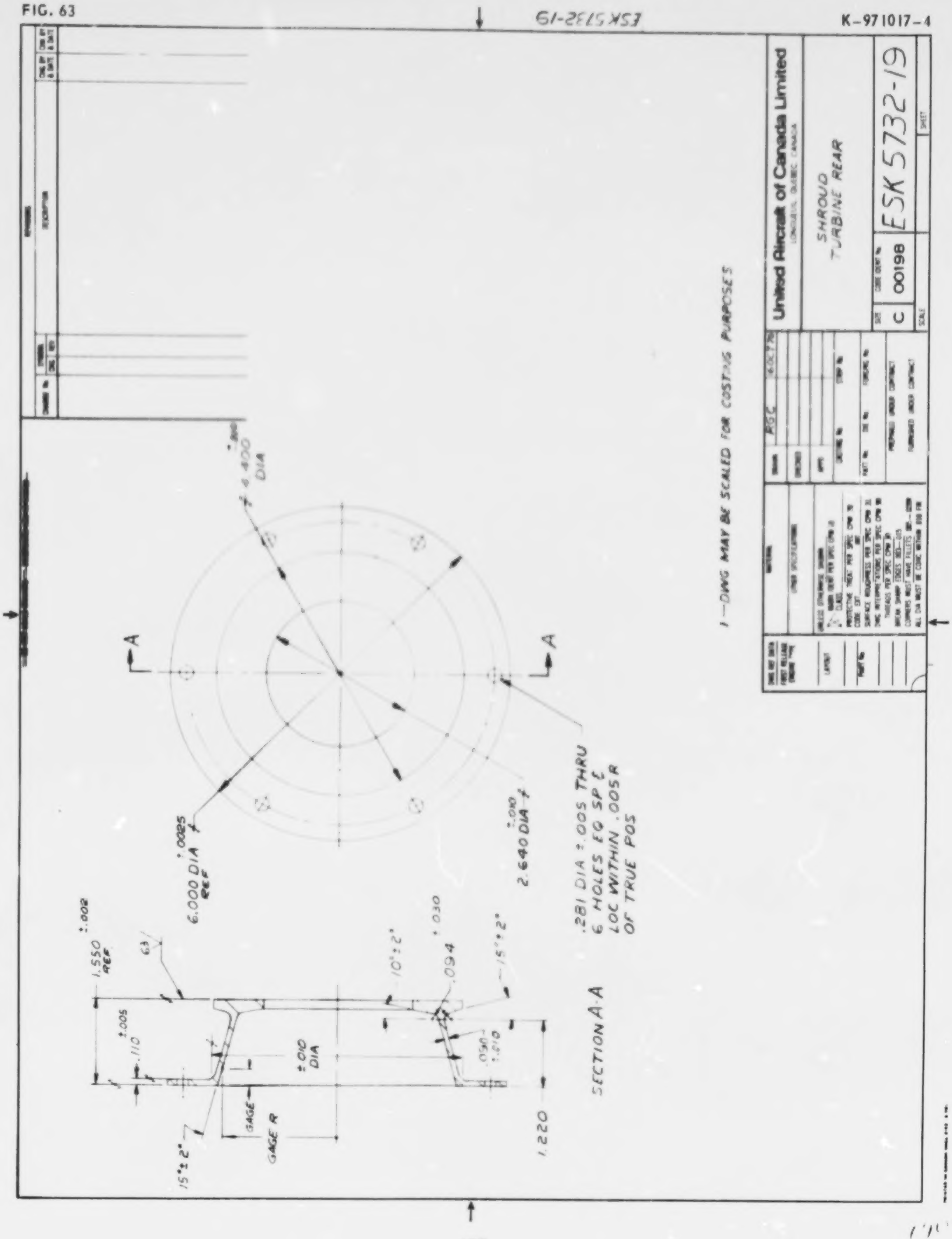

United Aircraft of Canada Limited
TURBINE - NOZZLE
F
00198
ESK 5732-11
SECTIONS A-A AND B-B
SCALE 2/1
PLANE [A] REPRESENTS ARITHMETICAL AVERAGE THRU 3 POINTS (C)
FACE [B]
4.900 ±.010 DIA
2.380 ±.010
GAGE
A
B
TRAILING EDGE OF EACH VANE MUST BE TANGENT WITH THIS DIA
SURFACE FINISH MAX ON ALL CAST SURFACES
DWG MAY BE SCALED FOR COSTING PURPOSES

FIG. 63

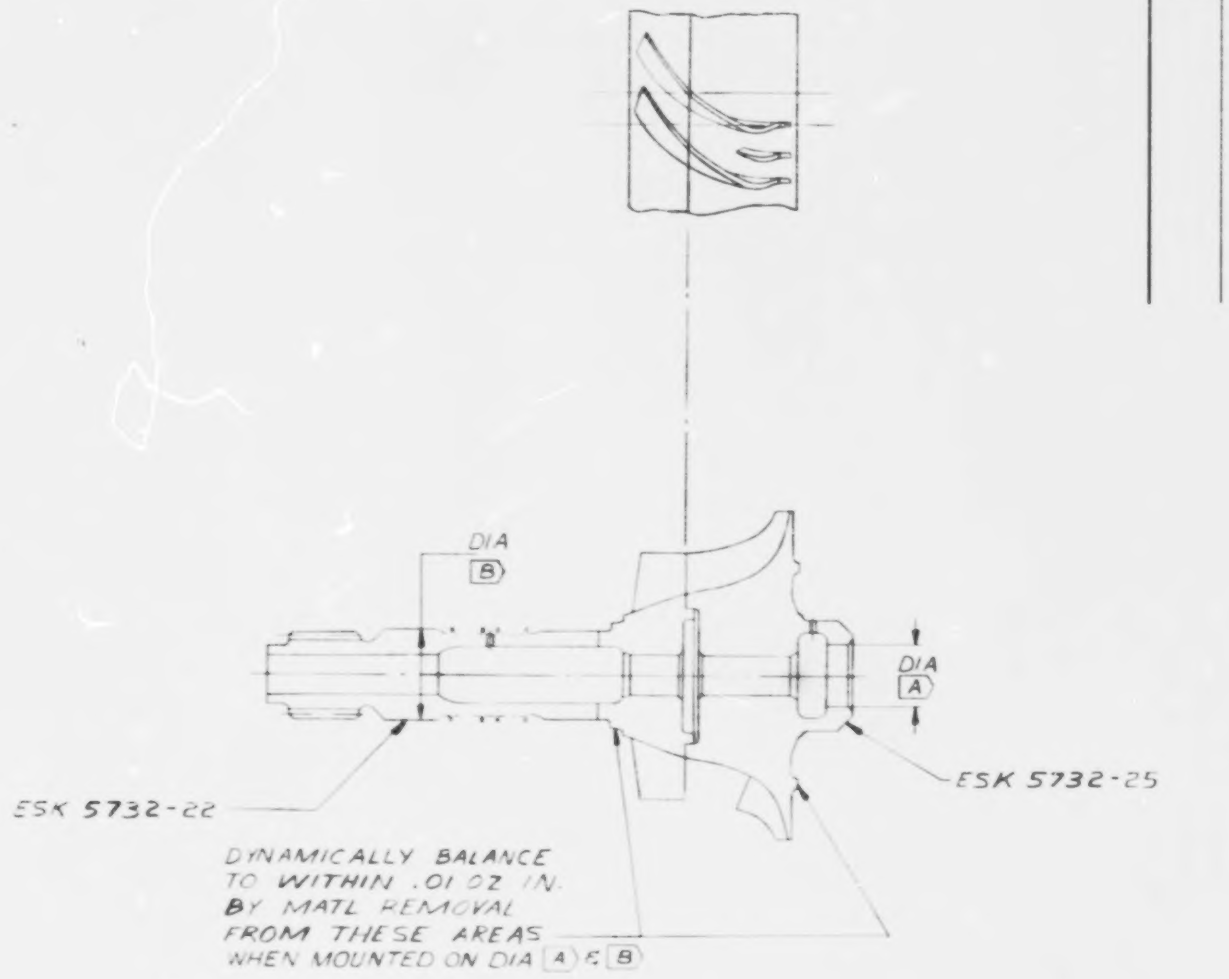
K-971017-4


United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA
SHROUD
TURBINE REAR
C
00198
ESK 5732-19
RGC
6 OCT 70
1—DWG MAY BE SCALED FOR COSTING PURPOSES
4.400 DIA ±.010
6.000 DIA ±.0025 REF
2.640 DIA ±.010
.281 DIA ±.005 THRU
6 HOLES EQ SP
LOC WITHIN .005 R
OF TRUE POS
A
A
SECTION A-A
1.550 ±.002 REF
63
.110 ±.005
±.010 DIA
10°±2°
.094 ±.030
15°±2°
.090 ±.010
1.220
GAGE
GAGE R
15°±2°

ESK 5732-21

— DO NOT SCALE DRAWING — DIMENSIONS ARE IN INCHES

CHANGE No	SYMBOL CHG	SYMBOL REV	REVISIONS DESCRIPTION	CHG BY & DATE	CHK BY & DATE



DYNAMICALLY BALANCE TO WITHIN .01 OZ IN. BY MATL REMOVAL FROM THESE AREAS WHEN MOUNTED ON DIA A & B

DWG MAY BE SCALED FOR COSTING PURPOSES

DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No

MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES 003—015
CORNERS MUST HAVE FILLETS 005—020R
ALL DIA MUST BE CONC WITHIN 010 FIR

DRAWN RGC 26OCT7
CHECKED
APPD
CASTING No STRIP No
PATT No DIE No FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT

United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA

ROTOR COMPRESSOR
ASSY

SIZE C
CODE IDENT No 00198
ESK 5732-21
SCALE
SHEET

FIG. 64

FIG. 65

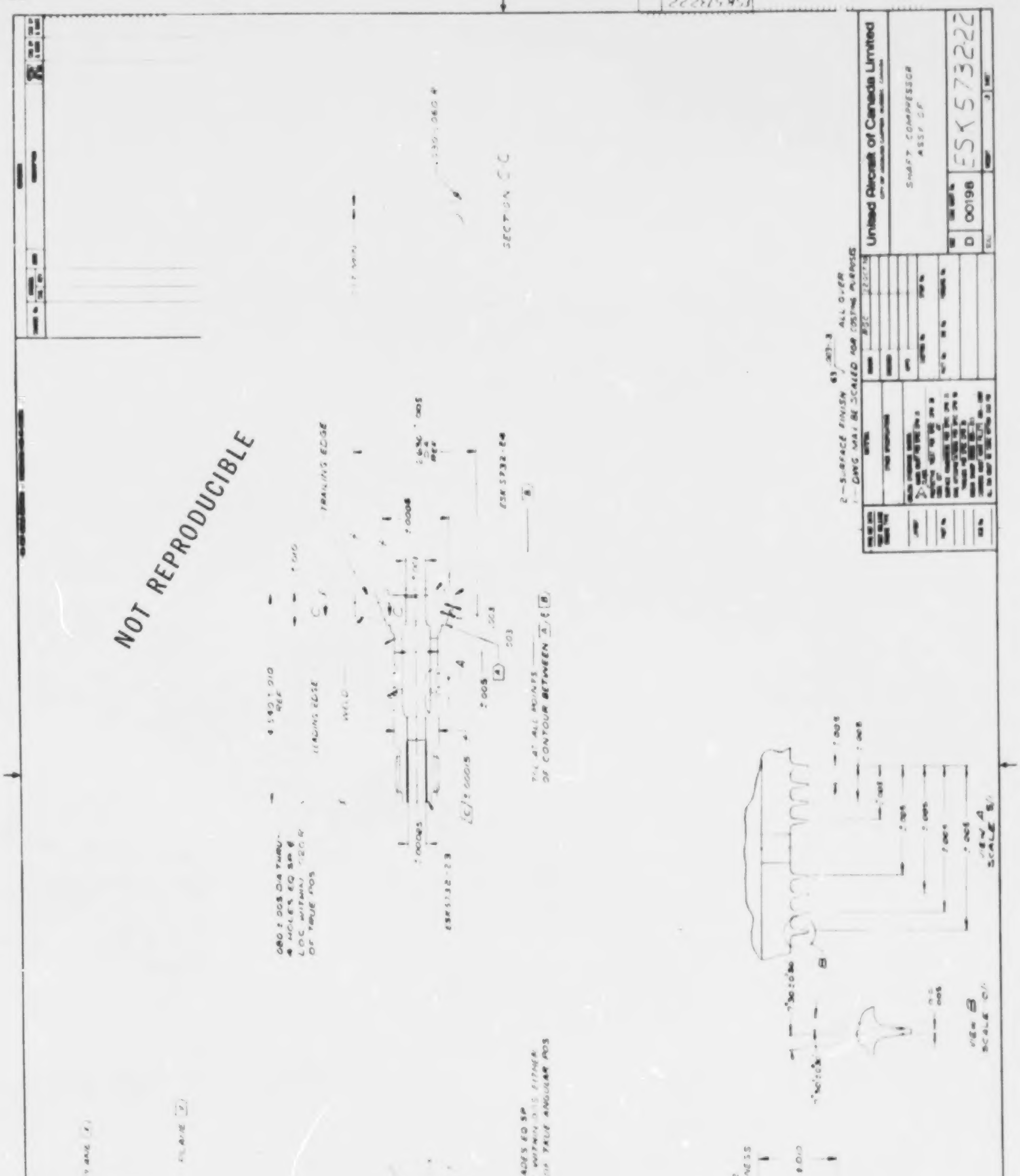

NOT REPRODUCIBLE
United Aircraft of Canada Limited
SHAFT COMPRESSOR ASSY OF
ESK573222
D
00198
SECTION C-C
LEADING EDGE
TRAILING EDGE
WELD
TOL AT ALL POINTS OF CONTOUR BETWEEN A & B
VIEW A SCALE 5/1
VIEW B SCALE 10/1
18 BLADES EQ SP
TYPICAL AIRFOIL SECTION
PLANE X
PLANE Y
ALL OVER

FIG. 66

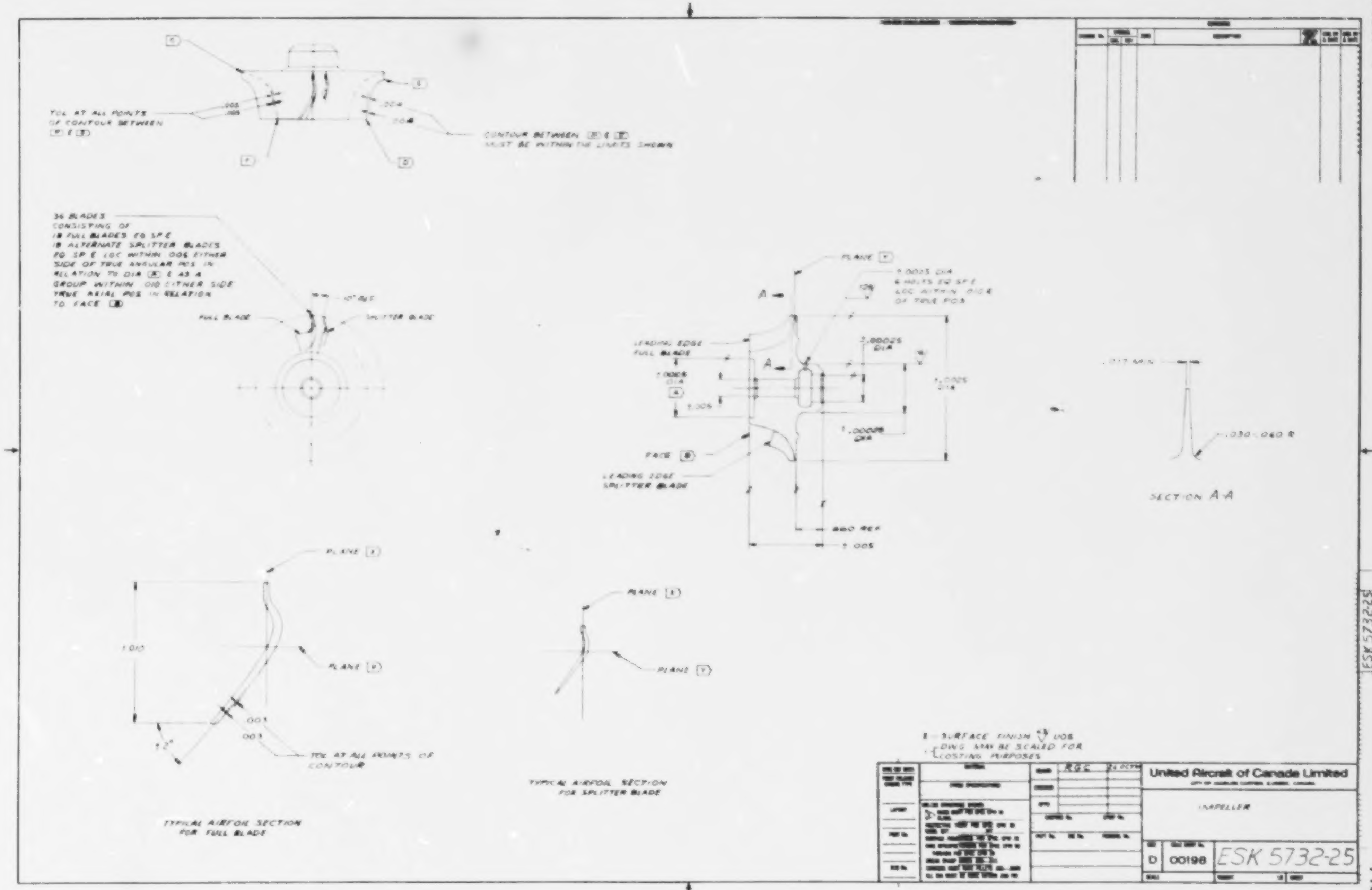

TOL AT ALL POINTS
OF CONTOUR BETWEEN
.005
.005
.004
.004
CONTOUR BETWEEN
MUST BE WITHIN THE LIMITS SHOWN
36 BLADES
CONSISTING OF
18 FULL BLADES EQ SP E
18 ALTERNATE SPLITTER BLADES
EQ SP E LOC WITHIN .005 EITHER
SIDE OF TRUE ANGULAR POS IN
RELATION TO DIA A E AS A
GROUP WITHIN .010 EITHER SIDE
TRUE AXIAL POS IN RELATION
TO FACE B
10° REF
FULL BLADE
SPLITTER BLADE
PLANE
±.0025 DIA
6 HOLES EQ SP E
LOC WITHIN .010 R
OF TRUE POS
A
LEADING EDGE
FULL BLADE
±.0005
DIA
A
±.005
±.00025
DIA
±.0005
DIA
±.00025
DIA
FACE B
LEADING EDGE
SPLITTER BLADE
.660 REF
±.005
.017 MIN
.030-.060 R
SECTION A-A
PLANE X
PLANE Y
1.010
.003
.003
TOL AT ALL POINTS OF
CONTOUR
TYPICAL AIRFOIL SECTION
FOR FULL BLADE
PLANE X
PLANE Y
TYPICAL AIRFOIL SECTION
FOR SPLITTER BLADE
SURFACE FINISH UOS
DWG MAY BE SCALED FOR
COSTING PURPOSES
RGC
United Aircraft of Canada Limited
IMPELLER
D
00198
ESK 5732-25

FIG. 67

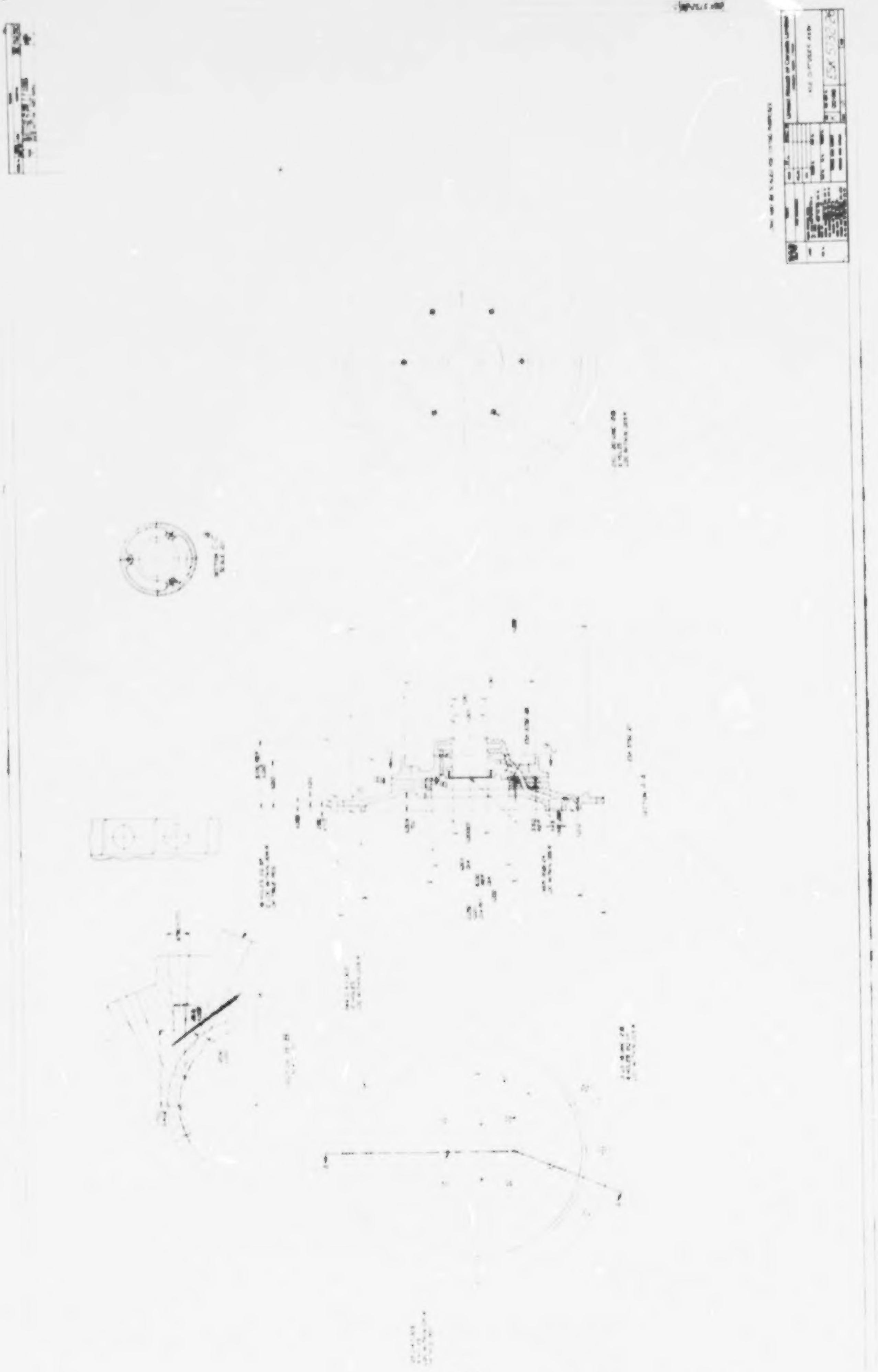


FIG. 68

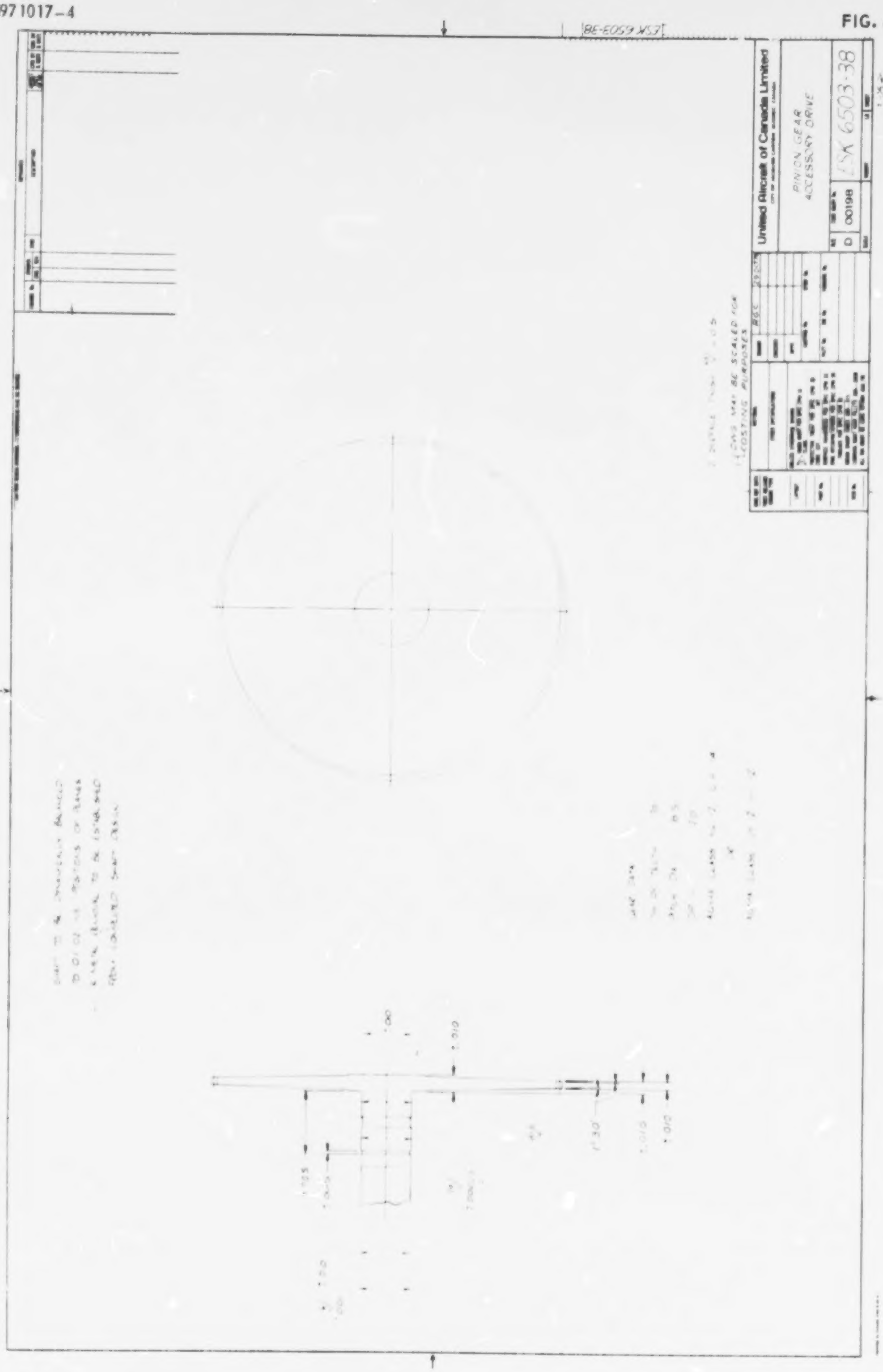

United Aircraft of Canada Limited
PINION GEAR
ACCESSORY DRIVE
D
00198
ESK 6503-38
DWG MAY BE SCALED FOR COSTING PURPOSES
ESK 6503-38

FIG. 69

K-971017-4

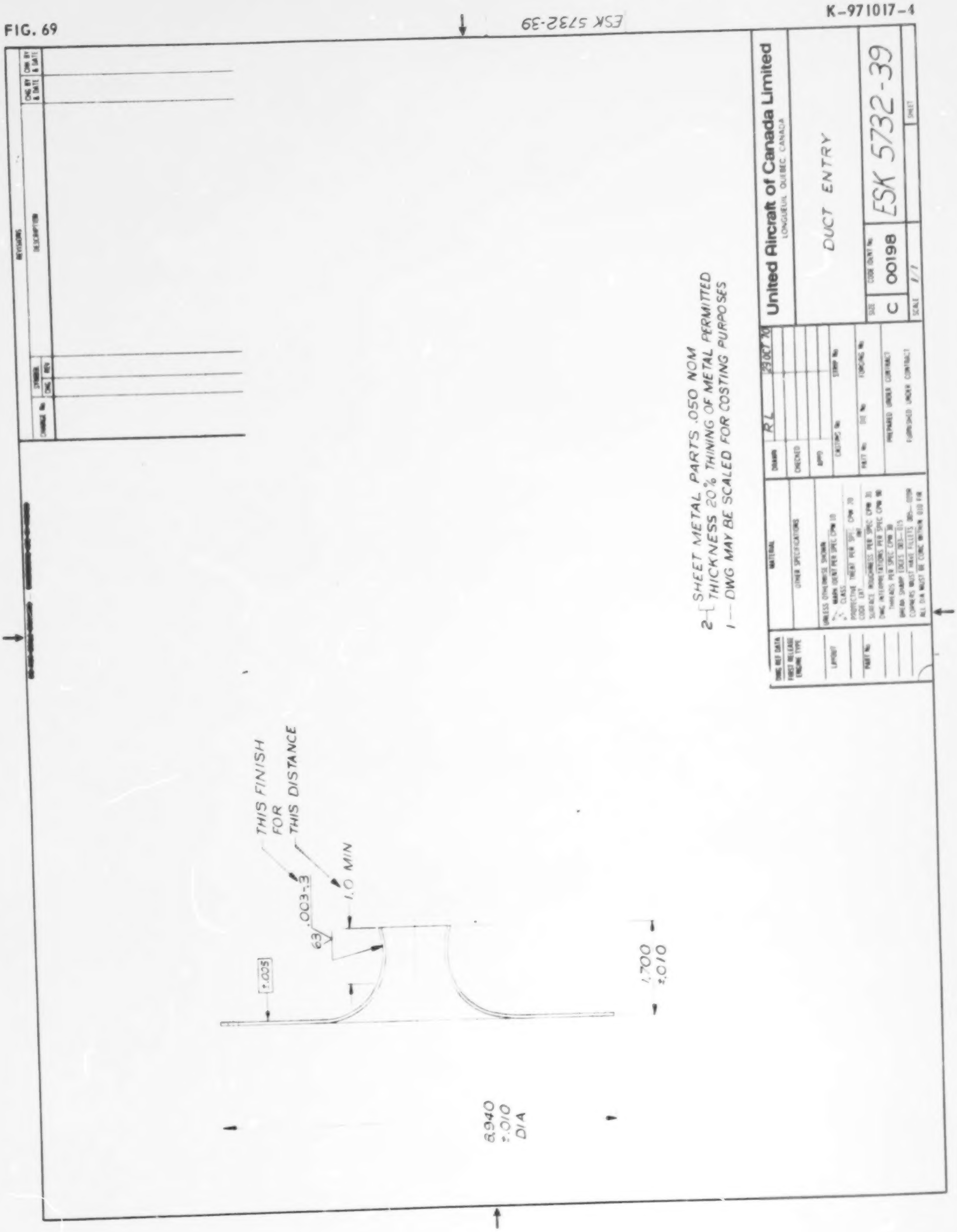

ESK 5732-39
United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA
DUCT ENTRY
DRAWN RL 29 OCT 70
CHECKED
APPD
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
SIZE C
CODE IDENT No. 00198
ESK 5732-39
SCALE 1/1
SHEET
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
REVISIONS
DESCRIPTION
2 — SHEET METAL PARTS .050 NOM THICKNESS 20% THINING OF METAL PERMITTED
1 — DWG MAY BE SCALED FOR COSTING PURPOSES
THIS FINISH FOR THIS DISTANCE
63 .003-.3
1.0 MIN
±.005
1.700 ±.010
8.940 ±.010 DIA

FIG. 70

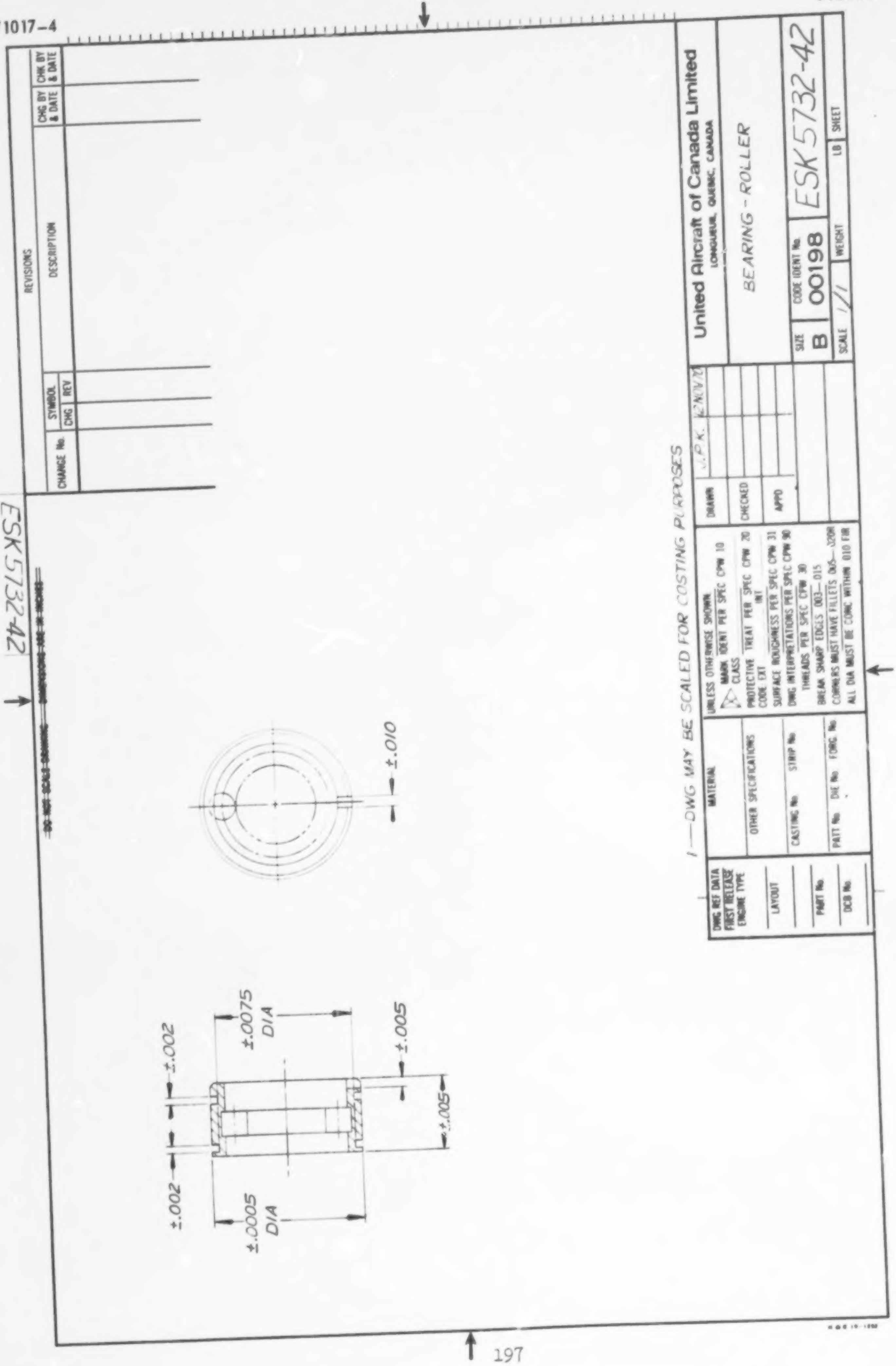

ESK5732-42
REVISIONS
CHANGE No.
SYMBOL CHG REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
DO NOT SCALE DRAWING DIMENSIONS ARE IN INCHES
±.002
±.0075 DIA
±.005
±.0005 DIA
±.005
±.010
1—DWG MAY BE SCALED FOR COSTING PURPOSES
United Aircraft of Canada Limited
LONGUEUIL, QUEBEC, CANADA
BEARING - ROLLER
DRAWN J.P.K. 12NOV72
CHECKED
APPD
SIZE B
CODE IDENT No. 00198
ESK5732-42
SCALE 1/1
WEIGHT
LB
SHEET
DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No
DCB No
MATERIAL
OTHER SPECIFICATIONS
CASTING No
STRIP No
PATT No
DIE No
FORG No
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
PROTECTIVE TREAT PER SPEC CPW 20
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003-.015
CORNERS MUST HAVE FILLETS
ALL DIA MUST BE CONC WITHIN .010 FIR

FIG. 71

United Aircraft of Canada Limited
GEAR REDUCTION
D
00198
ESK 5732-45
DWG MAY BE SCALED FOR COSTING PURPOSES
NOT REPRODUCIBLE
GEAR DATA

FIG. 72

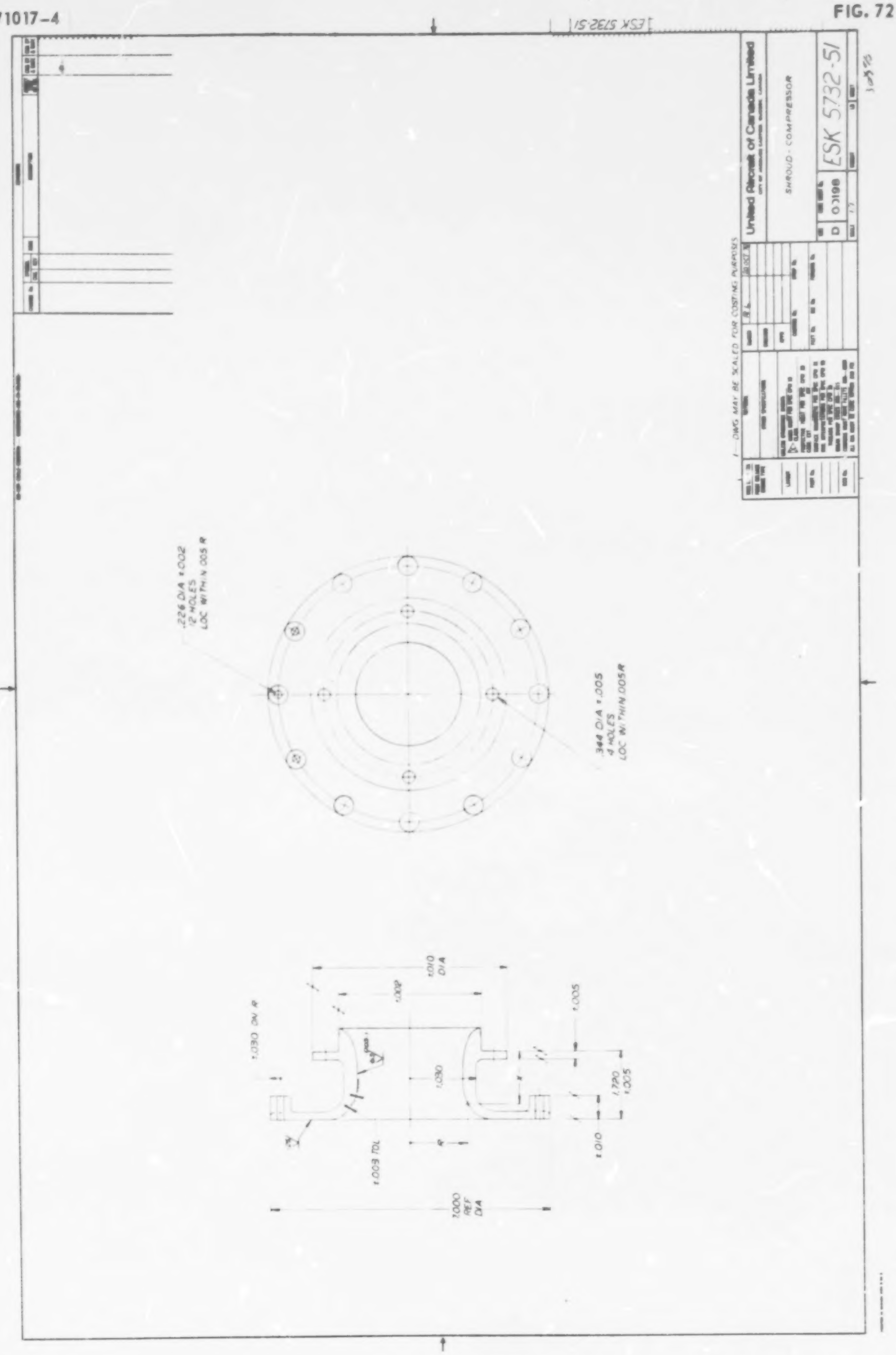

ESK 5732-51
United Aircraft of Canada Limited
SHROUD - COMPRESSOR
D 03198
ESK 5732-51
1—DWG MAY BE SCALED FOR COSTING PURPOSES
.226 DIA ±.002
12 HOLES
LOC WITHIN .005 R
.344 DIA ±.005
4 HOLES
LOC WITHIN .005 R
±.030 ON R
±.010
DIA
±.002
±.030
±.005
1.720
±.005
±.010
±.003 TOL
7.000
REF
DIA

FIG. 73

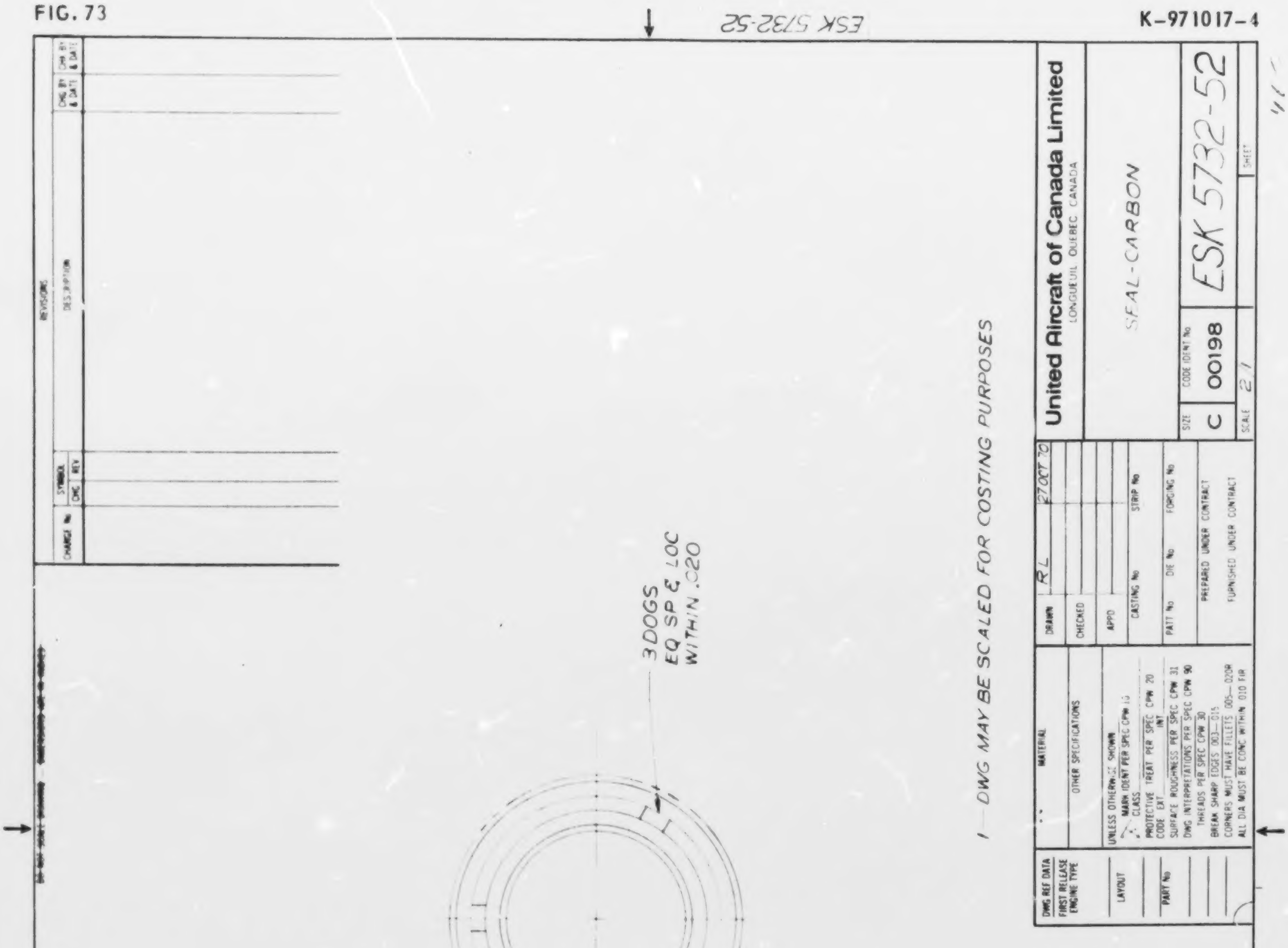

ESK 5732-52
United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA
SEAL-CARBON
SIZE C
CODE IDENT No 00198
ESK 5732-52
SCALE 2/1
SHEET
DRAWN RL 27 OCT 70
CHECKED
APPD
CASTING No
STRIP No
PATT No
DIE No
FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 13
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003-.015
CORNERS MUST HAVE FILLETS .005-.020R
ALL DIA MUST BE CONC WITHIN .010 FIR
DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No
REVISIONS
CHANGE No
SYMBOL
CHG
REV
DESCRIPTION
CHG BY & DATE
CK BY & DATE
1—DWG MAY BE SCALED FOR COSTING PURPOSES
3 DOGS
EQ SP & LOC
WITHIN .020
1.9375-20 UN-2A
THESE FACES
MUST BE PARALLEL
WITHIN .001
CASING TEMP
500°F MAX
P2 OIL
±.010
±.005
±.0005
±.010
±.010
±.005 2 SLOTS
EQ SP & LOC
WITHIN .010
AIR P1

FIG. 74

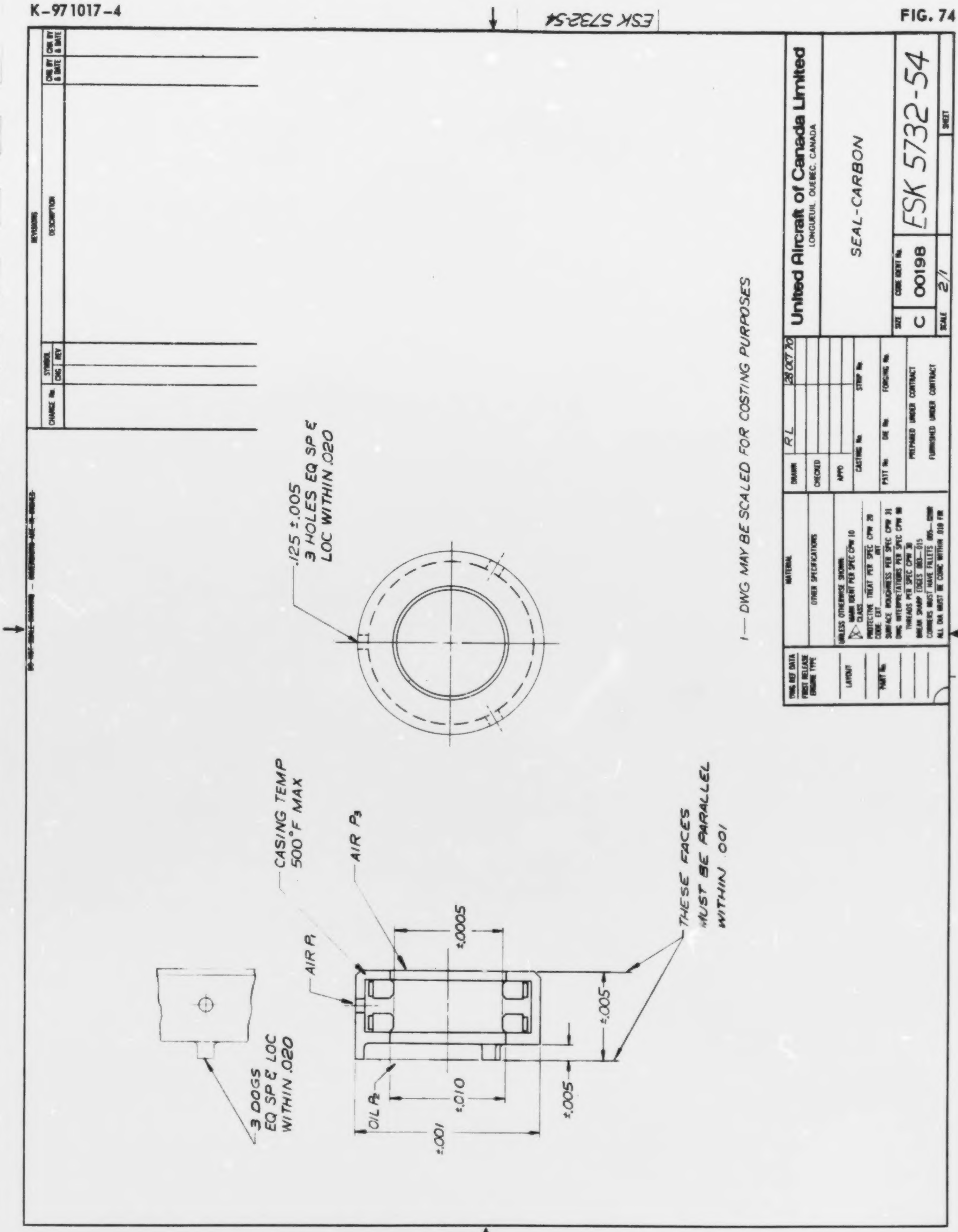
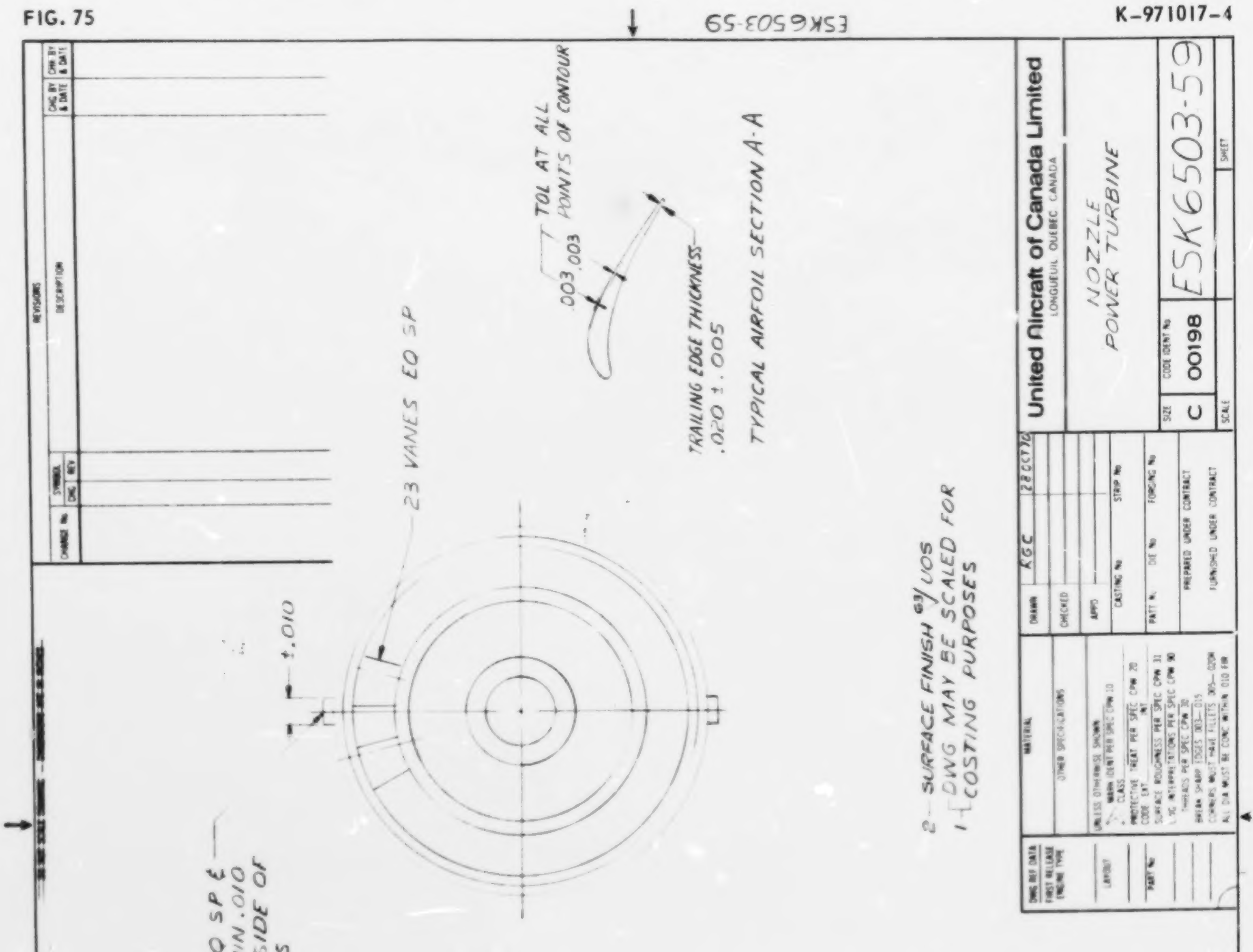
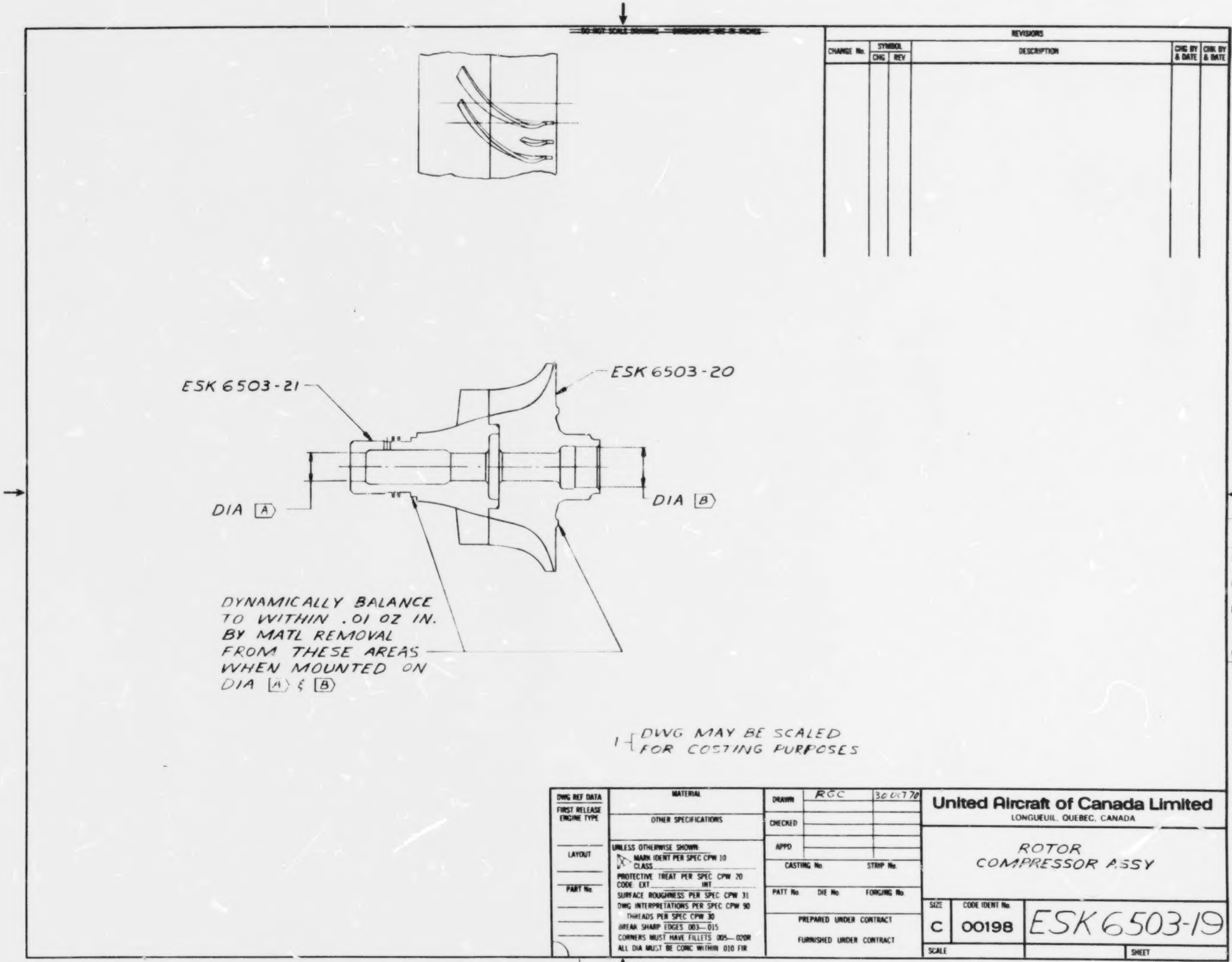

ESK 5732-54
REVISIONS
CHANGE No.
SYMBOL
CHG
REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
.125 ±.005
3 HOLES EQ SP ≰
LOC WITHIN .020
1— DWG MAY BE SCALED FOR COSTING PURPOSES
CASING TEMP
500°F MAX
AIR P3
AIR P1
±.0005
THESE FACES
MUST BE PARALLEL
WITHIN .001
±.005
±.010
OIL P2
±.001
±.005
3 DOGS
EQ SP ≰ LOC
WITHIN .020
United Aircraft of Canada Limited
LONGUEUIL, QUEBEC, CANADA
SEAL-CARBON
DRAWN
RL
28 OCT 70
CHECKED
APPD
CASTING No.
STRIP No.
PATT No.
DIE No.
FORGING No.
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 28
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003-.015
CORNERS MUST HAVE FILLETS .005-.030R
ALL DIA MUST BE CONC WITHIN .010 FIR
DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No.
SIZE
C
CODE IDENT No.
00198
ESK 5732-54
SCALE
2/1
SHEET
4995

FIG. 75


ESK6503-59
REVISIONS
CHANGE No.
SYMBOL
CHG
REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
2 DOGS EQ SP ℄
LOC WITHIN .010
EITHER SIDE OF
TRUE POS
23 VANES EQ SP
±.010
TOL AT ALL
POINTS OF CONTOUR
.003 .003
TRAILING EDGE THICKNESS
.020 ±.005
TYPICAL AIRFOIL SECTION A-A
2—SURFACE FINISH 63/ UOS
1—DWG MAY BE SCALED FOR
COSTING PURPOSES
±.005
±.005
±.005
±.005
.010
±.0005
DIA
±.001
DIA
±.001
DIA
32/
±.005
A
A
±.010
DIA
TOL AT ALL POINTS
OF CONTOUR ±.005
FOR THIS DISTANCE
REMAINDER MAY
INCREASE ±.010
125/
±.005 R
±.005
DIA
±.010
DIA
±.010
DIA
DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
L/G INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003-.015
CORNERS MUST HAVE FILLETS .005-.020R
ALL DIA MUST BE CONC WITHIN .010 FIR
DRAWN KGC 28 OCT 70
CHECKED
APPD
CASTING No
STRIP No
PATT No
DIE No
FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA
NOZZLE
POWER TURBINE
SIZE
C
CODE IDENT No
00198
ESK6503-59
SCALE
SHEET


DO NOT SCALE DRAWING — DIMENSIONS ARE IN INCHES
REVISIONS
CHANGE No.
SYMBOL
CHG
REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
ESK 6503-21
ESK 6503-20
DIA A
DIA B

DYNAMICALLY BALANCE TO WITHIN .01 OZ IN. BY MATL REMOVAL FROM THESE AREAS WHEN MOUNTED ON DIA A & B

1 { DWG MAY BE SCALED FOR COSTING PURPOSES

DWG REF DATA
FIRST RELEASE ENGINE TYPE
LAYOUT
PART No

MATERIAL

OTHER SPECIFICATIONS

UNLESS OTHERWISE SHOWN:
MARK IDENT PER SPEC CPW 10
CLASS ______
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT ______ INT ______
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES 003—015
CORNERS MUST HAVE FILLETS 005—020R
ALL DIA MUST BE CONC WITHIN 010 FIR

DRAWN: RGC 30 OCT 70
CHECKED:
APPD:
CASTING No
STRIP No
PATT No
DIE No
FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT

United Aircraft of Canada Limited
LONGUEUIL, QUEBEC, CANADA

ROTOR
COMPRESSOR ASSY

SIZE: C
CODE IDENT No: 00198
ESK 6503-19
SCALE
SHEET

FIG. 77

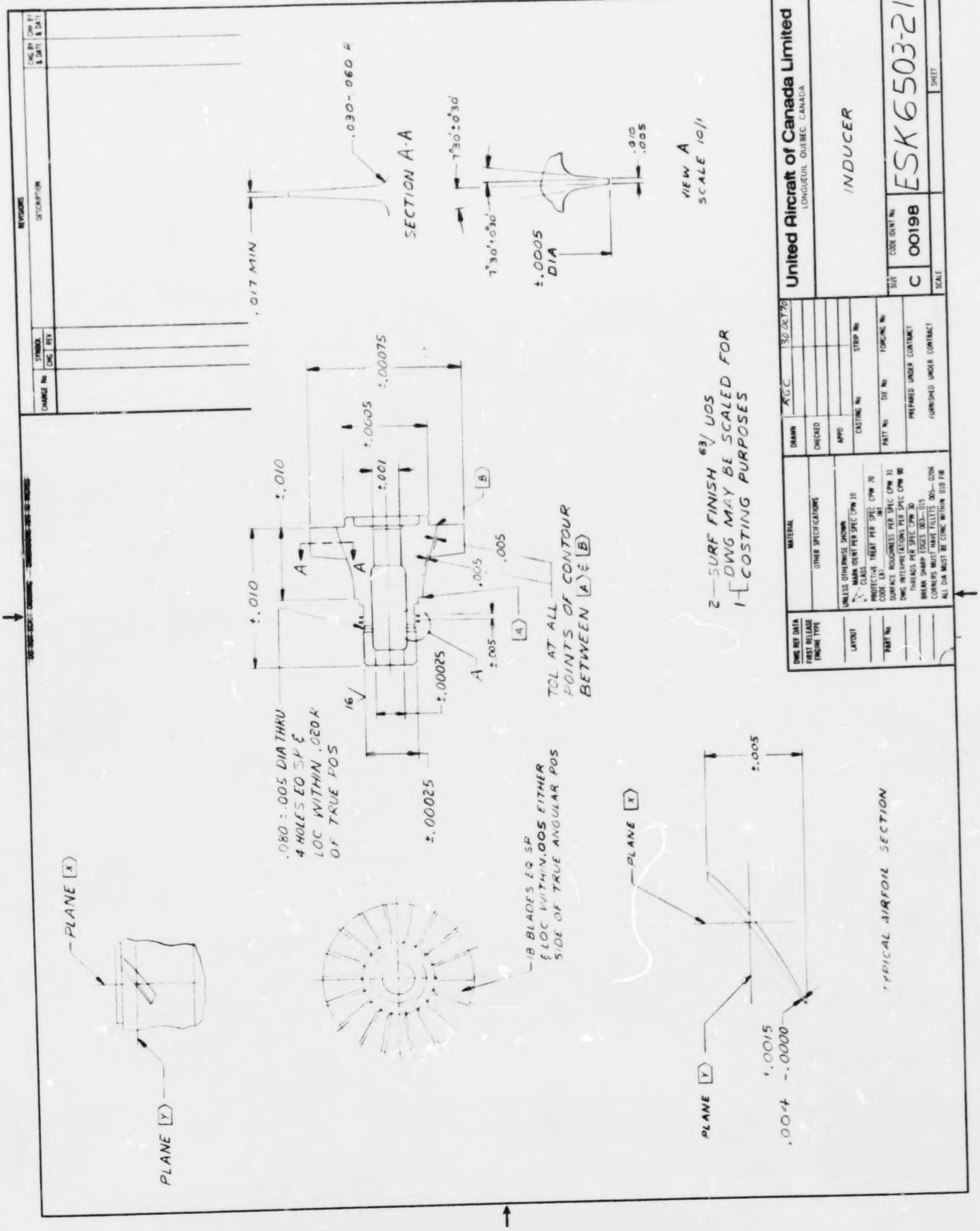

ESK6503-21
REVISIONS
CHANGE No
SYMBOL
DWG
REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
PLANE [X]
PLANE [Y]
-18 BLADES EQ SP
& LOC WITHIN .005 EITHER
SIDE OF TRUE ANGULAR POS
.080 ±.005 DIA THRU
4 HOLES EQ SP &
LOC WITHIN .020 R
OF TRUE POS
±.00025
16
±.00025
A
±.005
[A]
±.010
A
A
.005
.005
±.010
±.001
±.0005
[B]
±.00075
TOL AT ALL
POINTS OF CONTOUR
BETWEEN [A] & [B]
.017 MIN
.030-.060 R
SECTION A-A
7°30'±0°30'
7°30'±0°30'
±.0005
DIA
.010
.005
VIEW A
SCALE 10/1
2—SURF FINISH 63 UOS
1—DWG MAY BE SCALED FOR
COSTING PURPOSES
PLANE [X]
±.005
PLANE [Y]
+.0015
-.0000
.004
TYPICAL AIRFOIL SECTION
DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 50
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003-.015
CORNERS MUST HAVE FILLETS .005-.020R
ALL DIA MUST BE CONC WITHIN .010 FIR
DRAWN RGC 30 OCT 70
CHECKED
APPD
CASTING No
STRIP No
PATT No
DIE No
FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA
INDUCER
SIZE
C
CODE IDENT No
00198
ESK6503-21
SCALE
SHEET

FIG. 78

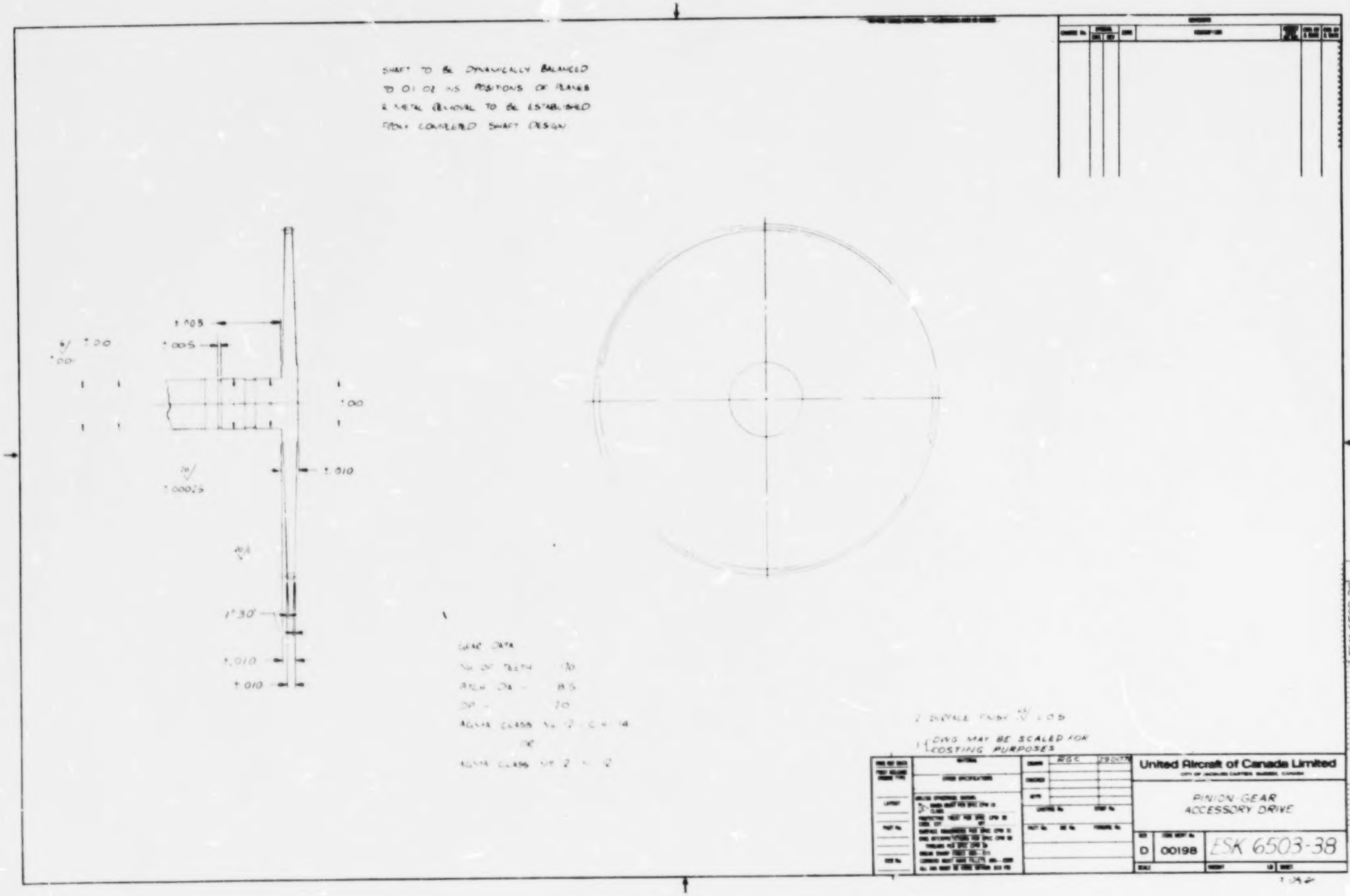
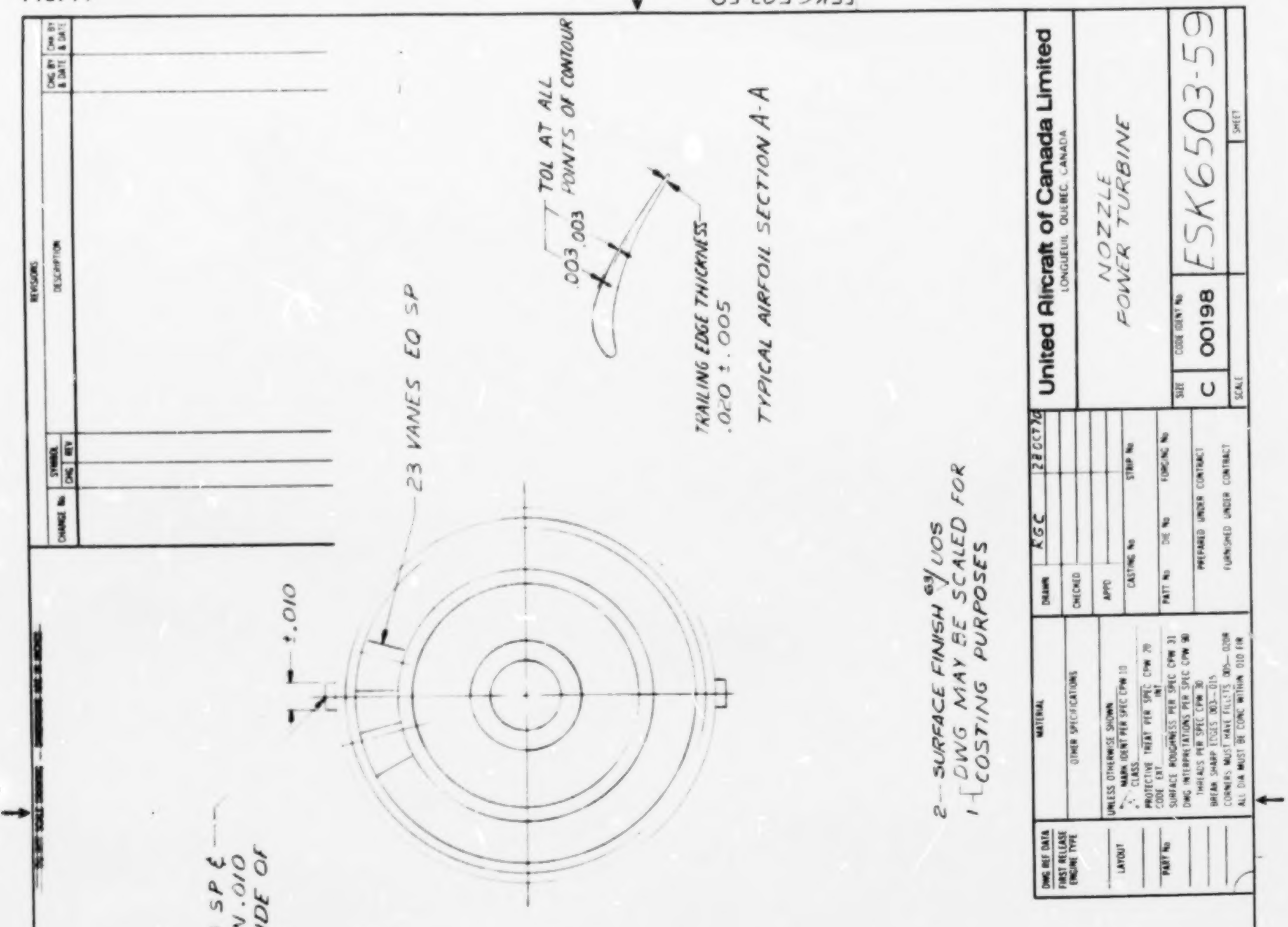

SHAFT TO BE DYNAMICALLY BALANCED
TO .01 OZ INS. POSITIONS OF PLANES
& METAL REMOVAL TO BE ESTABLISHED
FROM COMPLETED SHAFT DESIGN
±.005
±.005
±.00
±.00
±.010
±.00025
1°30'
±.010
±.010
GEAR DATA
NO OF TEETH 170
PITCH DIA - 8.5
DP - 20
AGMA CLASS No 12 - C 4 14
OR
AGMA CLASS No 2 × 12
SURFACE FINISH 32/ LOB
DWG MAY BE SCALED FOR COSTING PURPOSES
RGC
29 OCT 71
United Aircraft of Canada Limited
PINION-GEAR
ACCESSORY DRIVE
D
00198
ESK 6503-38

FIG. 79


REVISIONS
CHANGE No
SYMBOL
CHG
REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
23 VANES EQ SP
TOL AT ALL POINTS OF CONTOUR
.003 .003
TRAILING EDGE THICKNESS .020 ±.005
TYPICAL AIRFOIL SECTION A-A
±.010
2 DOGS EQ SP ℄ LOC WITHIN .010 EITHER SIDE OF TRUE POS
2— SURFACE FINISH 63√ UOS
1— DWG MAY BE SCALED FOR COSTING PURPOSES
±.005
±.005
±.0005 DIA
±.001 DIA
±.001 DIA
32√
±.005
±.005
±.005
±.010
A
A
±.010 DIA
125√
±.015 R
±.005 DIA
±.010 DIA
±.010 DIA
TOL AT ALL POINTS OF CONTOUR ±.005 FOR THIS DISTANCE REMAINDER MAY INCREASE ±.010
DWG REF DATA
FIRST RELEASE
ENGINE TYPE
LAYOUT
PART No
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 50
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003-.015
CORNERS MUST HAVE FILLETS .005-.020R
ALL DIA MUST BE CONC WITHIN .010 FIR
DRAWN
KGC
28 OCT 76
CHECKED
APPD
CASTING No
STRIP No
PATT No
DIE No
FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA
NOZZLE
POWER TURBINE
SIZE
C
CODE IDENT No
00198
ESK6503-59
SCALE
SHEET

ESK 6503-61

FIG. 80

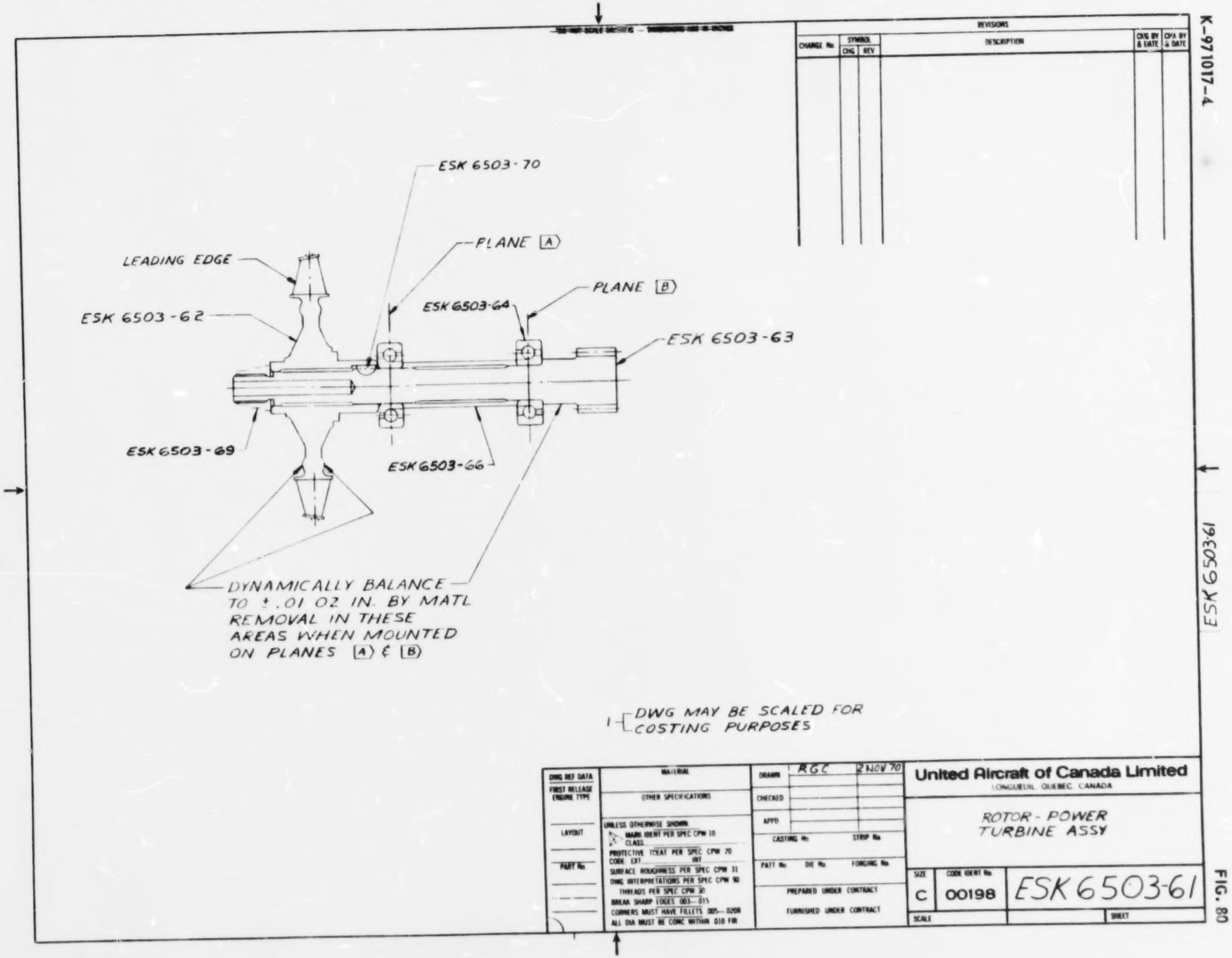
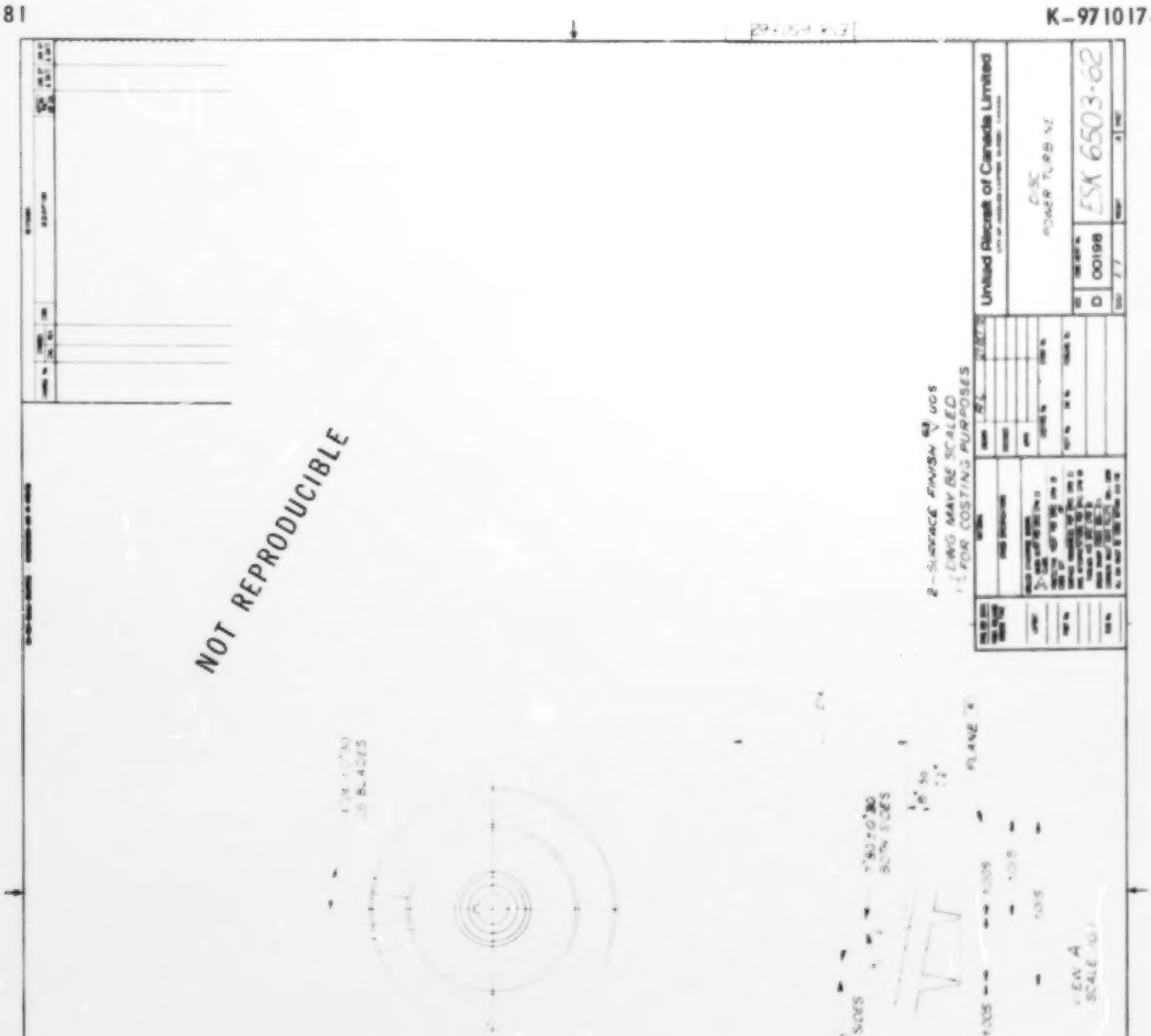
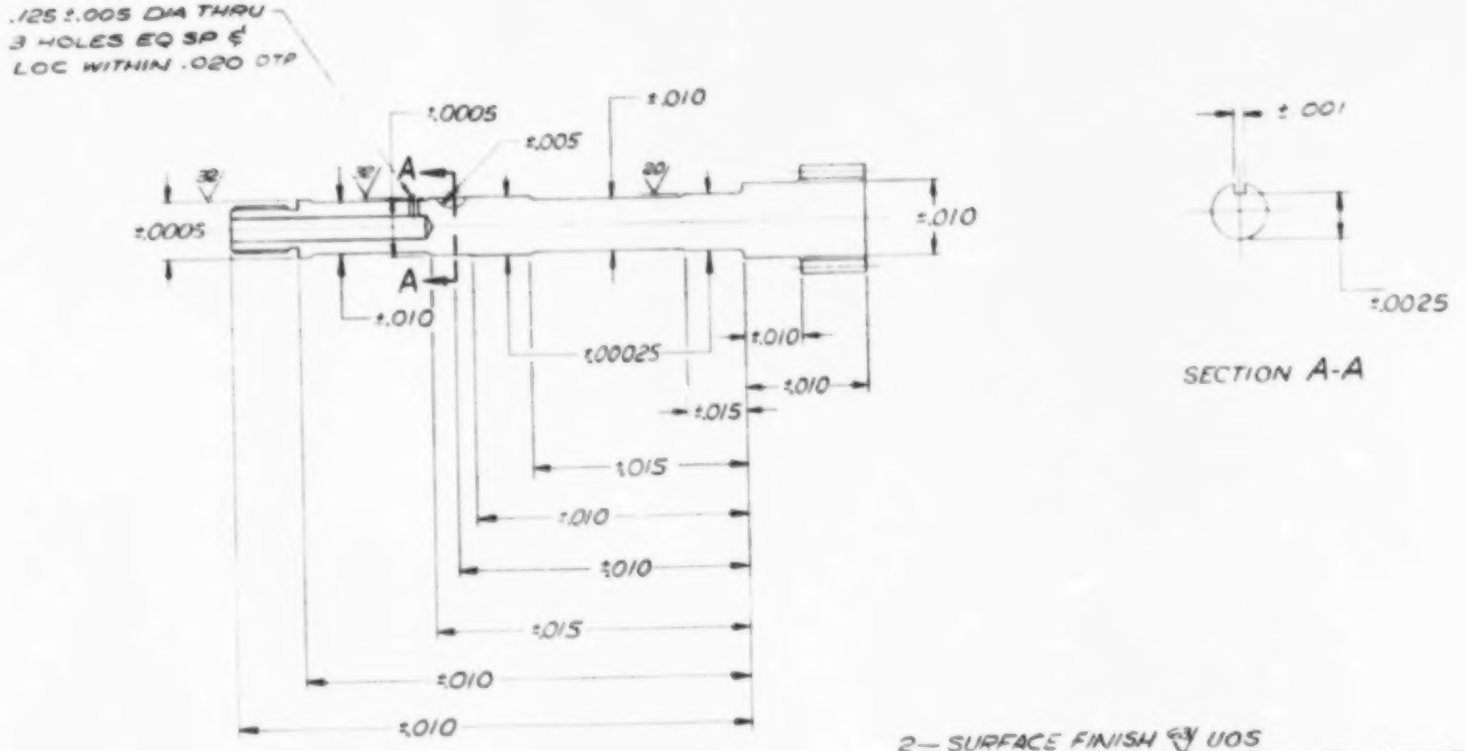

REVISIONS
CHANGE No
SYMBOL
CHG
REV
DESCRIPTION
CHG BY & DATE
CHK BY & DATE
ESK 6503-70
PLANE A
LEADING EDGE
PLANE B
ESK 6503-64
ESK 6503-62
ESK 6503-63
ESK 6503-69
ESK 6503-66
DYNAMICALLY BALANCE TO ±.01 OZ IN. BY MATL REMOVAL IN THESE AREAS WHEN MOUNTED ON PLANES A & B
1 DWG MAY BE SCALED FOR COSTING PURPOSES
DWG REF DATA
FIRST RELEASE ENGINE TYPE
LAYOUT
PART No
MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 90
THREADS PER SPEC CPW 30
BREAK SHARP EDGES 003—015
CORNERS MUST HAVE FILLETS 005—020R
ALL DIA MUST BE CONC WITHIN 010 FIR
DRAWN RGC 2 NOV 70
CHECKED
APPD
CASTING No
STRIP No
PATT No
DIE No
FORGING No
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT
United Aircraft of Canada Limited
LONGUEUIL QUEBEC CANADA
ROTOR - POWER TURBINE ASSY
SIZE
C
CODE IDENT No
00198
ESK 6503-61
SCALE
SHEET

FIG. 81


United Aircraft of Canada Limited
DISC
POWER TURBINE
ESK 6503-62
D
00198
NOT REPRODUCIBLE
2—SURFACE FINISH 63 UOS
1—DWG MAY BE SCALED
FOR COSTING PURPOSES
VIEW A
LEADING EDGE
TYPICAL AIRFOIL SECTION
PLANE
BOTH SIDES
BLADES
A

CHANGE No.	SYMBOL CHG	SYMBOL REV	DESCRIPTION	CHG BY & DATE	CHK BY & DATE


.125 ±.005 DIA THRU
3 HOLES EQ SP ⊄
LOC WITHIN .020 OTP
±.0005
±.005
±.010
32
32
20
A
A
±.0005
±.010
±.010
±.00025
±.010
±.010
±.015
±.015
±.010
±.010
±.015
±.010
±.010
±.001
±.0025
SECTION A-A

2— SURFACE FINISH $\overset{63}{\vee}$ UOS
1— DWG MAY BE SCALED FOR COSTING PURPOSES

GEAR DATA

NO. OF TEETH 20
DP 20
PD 1.0
AGMA CLASS NO 12 - CH-14
OR
AGMA CLASS NO 12 - N - 12

DWG REF DATA
FIRST RELEASE ENGINE TYPE
LAYOUT
PART No.

MATERIAL
OTHER SPECIFICATIONS
UNLESS OTHERWISE SHOWN
MARK IDENT PER SPEC CPW 10
CLASS
PROTECTIVE TREAT PER SPEC CPW 20
CODE: EXT INT
SURFACE ROUGHNESS PER SPEC CPW 31
DWG INTERPRETATIONS PER SPEC CPW 10
THREADS PER SPEC CPW 30
BREAK SHARP EDGES .003—.015
CORNERS MUST HAVE FILLETS .005—.020R
ALL DIA MUST BE CONC WITHIN .010 FIR

DRAWN RL 28 OCT 70
CHECKED
APPD
CASTING No. STRIP No.
PATT No. DIE No. FORGING No.
PREPARED UNDER CONTRACT
FURNISHED UNDER CONTRACT

United Aircraft of Canada Limited
LONGUEUIL, QUEBEC, CANADA

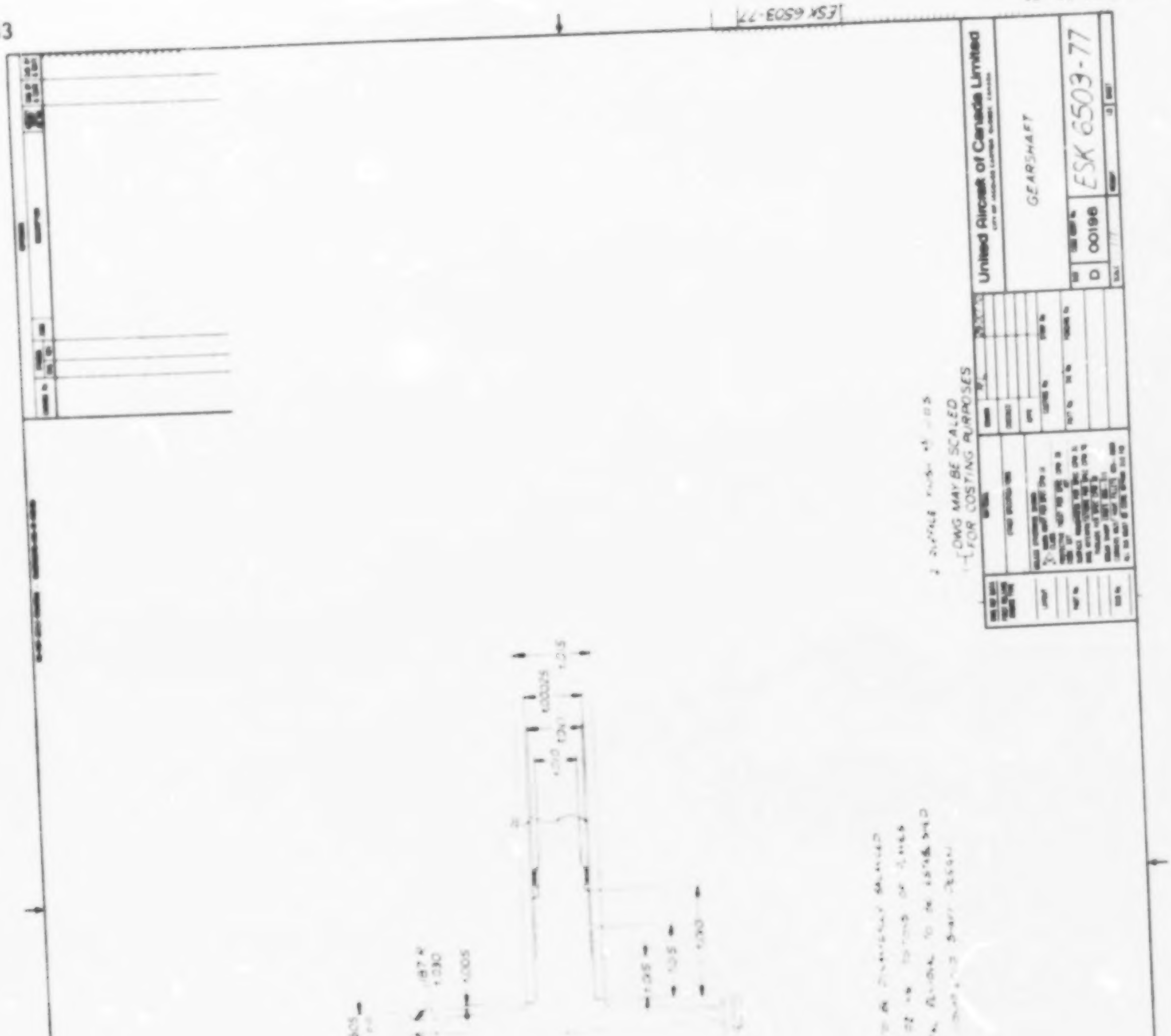
GEARSHAFT
POWER TURBINE

SIZE C
CODE IDENT No. 00198
ESK 6503-63
SCALE 1/1
SHEET

ESK 6503-63

FIG. 82

FIG. 83


ESK 6503-77
United Aircraft of Canada Limited
GEARSHAFT
D
00198
ESK 6503-77
DWG MAY BE SCALED FOR COSTING PURPOSES
GEAR DATA
NO OF TEETH 30
DP 20
PD 1.5